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Some Statistical Methods for Evaluation of Experimental Results

By E. W. PIKE,[1] NEWTON, MASS.

The statistician plays an important part in this machine age in which we live. Statistical examination of mass-production methods, for example, results in improvement, better quality control, and more uniformity of billions of dollars' worth of industrial products produced each year by repetitive processes. This paper describes the logic and methods used in evaluating and organizing equipment tests.

Introduction

IF any of you are wondering why an ex-astronomer employed by an electronics-design laboratory should be talking about statistics before a distinguished gathering of mechanical engineers, I can reply only by the rhetorical question "When did a man ever need an excuse to talk about his hobby?" I'm deeply grateful to the Power Test Codes Committee, however, for the opportunity to do so here. It is not only a pleasure but also an honor and a privilege.

In order to make clear what a statistician can contribute to this discussion, it is necessary to emphasize the obvious—in this case, that the purpose of these tests is to come to a conclusion on the basic question, "Does this equipment conform to the specifications?" The notebooks full of readings, the careful studies of instrument aberrations, are valuable only in so far as they are the bases for a conclusion on this primary question.

A statistician is a man who tries to be professional about drawing conclusions from facts. The great statistical teacher Snedecor has defined a statistician as "a skilled practitioner of the art of uncertain inference." In pursuing this profession or art of statistics, the statistican has developed a mathematical apparatus comparable in volume, subtlety, and range of difficulty to the subject of kinematics—including general relativity. In addition, he has wrestled with all the devils of epistemology and has taken many bad bruises in a struggle which is by no means concluded.

Viewed as a manufacturer of conclusions, the statistician shares many of the troubles familiar to all manufacturers. His raw materials are expensive, often in short supply, and subject to contamination from unidentifiable sources. His is strictly a custom product, no two alike, so that he must obtain and process a tremendous variety of raw materials. The unusual fact that his customers are, almost without exception, his raw-material suppliers is a mixed blessing. On the one hand, it encourages competition from "home industry." On the other hand, it enables him to realize very important savings by co-operating in the production of his raw facts. By taking only those observations which apply directly to the desired conclusion, and by avoiding through advance knowledge the most serious types of contamination, the value of the conclusions produced per dollar of test expense can often be increased by a factor of two or three. Factors of 10 or 20 have been recorded.

Statistician's Logic

There would be little point in trying to give a course in statistical mathematics in the next 15 min. There are good texts and good courses for anyone who wishes to study this field. I do wish to present, very briefly, the kind of logic which a statistician uses in drawing conclusions, and to present the two main mechanisms by which statistical help can improve the process of generating facts.

The objective of power-plant equipment tests is, once again, to reach a conclusion concerning the question, "Does this equipment conform to the specifications?" In reducing the data from the tests on any given equipment, however, it is not correct to speak in terms of the "probability" that this given equipment conforms. It either conforms, or it does not, and this is a physical condition about which there is no uncertainty. It is the test procedure which is uncertain. No finite amount of testing will decide unerringly whether any given equipment conforms or not. Correct logic, then, requires that we investigate the probability that some given test procedure will lead to a correct conclusion. If this probability is too low, then this procedure must be expanded to include additional readings.

Suppose now that we have a set of actual test results. For any specific postulated characteristics of the equipment, it is a straightforward problem (though sometimes laborious) to compute the probability that applying our actual test procedure to the postulated characteristics would produce the actually observed results. In this calculation, the known characteristics of the instruments and of the observers should be taken into account.

As a simple illustration of this idea, suppose I go to one of these penny-in-the-slot gum machines in the subway, and for two cents I get two red gum balls. Now, if I assume that there are three times as many red gum balls as white gum balls, I can calculate what the chance of drawing two red balls is. (Of course, the man who filled the machine might have crossed me up by putting all red on the bottom of the jar, and white on top, but these are the chances one must take. More about this later.) Again, I might assume that there were only two red balls in the machine, the rest being green, black, and yellow. Again, I can calculate the chance that I would get two red balls from this situation, and this time it would be pretty small.

In much the same way, but on a much larger scale, the probability of getting, by chance, the actually observed readings can be calculated, for a variety of postulated characteristics, both inside and outside of the specification range. If the probability of getting the actual readings is relatively large when the equipment is within specifications, and is relatively small when the equipment is postulated to be outside the specification limits, then one can conclude that either the equipment is within specifications, or we have been the victim of an improbable event. If the probability, that equipment whose characteristics are outside the specification range will produce the observed readings, is never greater than 1 per cent of the probability that equipment as specified will produce the same set of observed readings, then one con-

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cludes with a confidence of 99 per cent that the equipment conforms to specifications.

Statistical Experiment

This discussion leads directly to the idea of statistical experiment design. On the basis of his calculations, the statistician can determine which measurements will respond most sensitively to the differences between equipment which meets specifications, and equipment which does not. In other words, he can plan the experiment so that only those observations which contribute directly to the desired conclusions are taken. Quantities which are insensitive to the differences between acceptable and rejectable equipment need not be measured. Furthermore, the number of observations necessary to reach some specified conclusion with any desired degree of confidence can usually be estimated. Thus the cost of testing can be balanced against the cost of an incorrect conclusion; the degree of testing which leads to minimum total cost would be chosen, other things being equal.

In making these estimates, the statistican would make use of all the available information on the possible aberrations of the instruments and the personnel involved in the test. He would also make use of the best available engineering judgment as to the specific ways in which the equipment might be expected to deviate from specifications. The number of observations needed to decide with a given confidence between two specific alternatives is usually very much less than that needed to decide that some statement is true, excluding all possible alternatives. Thus the cost of testing may be reduced considerably, in many cases, without an appreciable increase in the cost of false conclusions, by testing primarily against the likely deviations from specification.

It is difficult to tell, by a casual examination, just how much might be saved in any particular case by careful planning of a test along statistical lines. Intuitive planning by intelligent engineers with long experience has led, in those cases where careful studies have been made, to widely variable results. Every story of incredibly bad planning can be matched with a story of superbly efficient planning, and vice versa. While I am not conversant with the latest results in this field, estimates of average improvements in the order of two to one are fairly common among statisticians.

A second, and probably even more important, contribution of the statistician to the process of testing is inherent in the fact that testing is a process. An acceptance test is a series of actions which are specified in advance, and then executed in reality. These actions are usually repetitive—read a thermometer every 5 min, or maintain a constant feed-water temperature for several hours.

Now mass production is exactly such a series of repetitive operations, specified in advance and then executed in reality. Because of its economic importance, mass-production processes have been studied in great detail by almost every kind of expert there is, including statisticians. As a result, a great deal is known about the characteristics of repetitive operations. The experimental fact which is of importance to us here is that no repetitive operation is ever really repeated exactly. There are always changes with time, sometimes irregular, but usually systematic; they arise from the gradual and unconscious formation of habits by operators, from variations in weather, in line voltage, from the wear of tools and machines, and from innumerable other causes.

These unsuspected changes can be detected by statistical examination of the output of the process. Furthermore, this statistical examination of the output often gives very helpful clues as to the form of the variation, and through this to the causes of variation. These causes can be identified and eliminated, one by one, with the result that the product of the process is much more uniform than it was originally.

Under the name of "statistical quality control," this procedure for improving and controlling and measuring the uniformity of a repetitive process is applied to billions of dollars' worth of industrial products every year, and the estimated savings, over the manufacturing methods of the 1930's, say, are in the same order of magnitude.

Some tens of thousands of industrial processes have been studied by quality-control engineers, without a single report of a process which was actually stable on first examination.

The same results have been found, on a smaller scale, when the process of testing is examined in the same way. Several hundred engineering tests have been examined (primarily at Bell Telephone Laboratories and at the National Bureau of Standards, but in many other organizations as well) both in the laboratory and on the test floor. Up to now, no one has reported a test which was stable on first examination. I have examined several of the great classic experiments of physics, from the point of view of their stability. Every one for which there was enough data to form a judgment had been out of control.

There is some evidence that a series of 500 star transits, made in 1833 by the great German astronomer Bessel, is actually stable and in control. This series is unusually important because it has been quoted for more than a century as proof that naturally occurring errors are distributed according to the so-called "normal law." This occurs only when the observing process is in control, so that apparently an exceptional case was taken, accidentally, as representative of the class.

The causes of unstable behavior in test runs are as diverse as the tests themselves. They fall into the same general classes as the causes of variation in production—habit formation, wear and slippage, variations in the weather, in raw materials, in power or water supply, etc. The methods of finding and identifying the causes of this unstable behavior are very similar to those used in the statistical quality control of production processes. The numerical methods are somewhat different because of the shorter runs encountered.

All of these methods for detecting and identifying the causes of instability depend on the examination of sequences of readings which are nominally constant, and which, in fact, should vary randomly, over a small range, around a given value. Consider, as an extreme example, the following sequence of observed temperatures of a feedwater intake, say, which should be at 110 F: 110.2, 109.7, 109.6, 110.1, 110.3, 109.9, 112.3, 111.9, 112.1, 111.7, 112.4, 112.1. It is quite clear that something happened between the sixth and seventh readings. More acute examination, with the aid of computations, and precalculated tables, will serve to find individual gross blunders, steps (like the one in the sequence given), ramps, cycles, undue variation, erratic "jumping" between two alternate states, and many other kinds of changes which will have meaning to an engineer familiar with the particular kinds of trouble that his equipment might produce.

Conclusions

Summing up this ramble through the tangled borderland between testing and statistics, the function of the statistician can be described as improving the efficiency with which conclusions can be drawn from testing. This improvement can come from the use of specialized mathematical methods on existing test results. Much more, however, can be expected from close co-operation between the statistician and the test engineer, resulting in a test which takes no useless or ineffective facts, and which is, in fact, the test which was planned in advance.

The ideal situation, of course, is the combination of test engineer and statistician in one person. This is a fairly common occurrence in engineering at the present time; usually an interested engineer learns enough statistics to act quite effectively as his

own statistical consultant, and not a few have become outstanding professional statisticians. A word of warning, though—statistics is a difficult subject to master by self-study, and graduate study at a university, if it can possibly be arranged, is almost always well worth while in terms of time and money saved.

REFERENCES

For the benfit of those who may wish to read further in this field, a few useful books are mentioned. There has been no attempt to cover the field.

"Introduction to Mathematical Statistics," by P. G. Hoel, John Wiley & Sons, Inc., New York, N. Y., 1947, is an excellent starting point for a general study of statistics.

"Design of Experiments," by R. A. Fisher, Oliver and Boyd, Ltd., Edinburgh, Scotland, 1937, is delightful reading, by a master.

"Statistical Methods in Research," by P. O. Johnson, Prentice-Hall, Inc., New York, N. Y., 1949, is an excellent guide to the simpler phases of practical statistics.

"An Introduction to Scientific Research," by E. Bright Wilson, Jr., McGraw-Hill Book Co., Inc., New York, 1952. A very fine general reference on when, why, and how to measure.

"Experimental Designs," by W. G. Cochran and G. M. Cox, John Wiley & Sons, Inc., New York, N. Y., 1950, covers the classic work on experimental design from the point of the experimenter.

"The Economic Control of Quality of a Manufactured Product," by W. A. Shewhart, D. Van Nostrand Co., Inc., New York, N. Y., 1934, while old, still gives the best presentation of the ideas involved in statistical quality control, although the texts of Juran and of Grant give more modern methods.

A series of articles by P. A. Olmstead, in *Industrial Quality Control*, vol. 9, November, 1952, and January, 1953, are the outstanding material on control of tests, although they require some background to follow.

Design of Power-Plant Tests to Insure Reliability of Results

By W. A. WILSON,[1] CAMBRIDGE, MASS.

This paper deals with the application of uncertainty analysis to the planning of tests. A technique for defining alternative features of test design and of choosing between them is discussed. It is shown that one automatically makes most of the important decisions with respect to test setup and procedure in the course of the uncertainty analysis.

Introduction

THIS paper discusses the problem of designing tests that will insure acceptably reliable results. This aim is related closely to other principal purposes of test plans, including the following:

1 Determination of required performance relationships.

2 Control of testing costs by the following:

(*a*) Judicious selections of place and mode of operation and of instrumentation.

(*b*) Smooth operational and data-gathering procedure and the economical sequencing of runs.

(*c*) Appropriate division of labor among the test crew.

(*d*) Avoidance of back-tracking by setting up an on-the-spot analysis to expose anomalous results.

3 Maximization of usefulness of data gathered by the following:

(*a*) Anticipation of operating ranges of most interest.

(*b*) Application of generalizing analysis and interpretation.

Merely to illuminate the single problem of securing an acceptable uncertainty interval in the result is enough of a problem. Inevitably, however, in discussing this limited objective, considerable light will be reflected upon these other aspects of test design.

The discussion is presented in the form of a "cookbook" procedure which is particularly applicable to the design of performance tests similar to those for which codes exist. Research engineers may be revolted by such a mechanical approach; and initiates, generally, may be impatient with the numerous statements of the obvious. However, the formula propounded to illustrate the application of uncertainty principles does not diminish the need for skill and judgment; it merely suggests how they may be applied in an orderly manner to an important problem which frequently receives too casual treatment. What is being advocated is development of a methodical approach to a problem usually handled intuitively.

Basic Procedure

The essential feature of the method is a series of uncertainty analyses based on procedures described by Kline and McClintock.[2] These analyses are to be made in advance of the tests. They are applied to alternative test designs and provide a means of rating the suitabilities of the hypothesized procedures on the basis of the uncertainties which would derive from their application. Both the technique for selecting the alternatives to be analyzed and the methods of the uncertainty analyses themselves are considered. Final selection of test design is not discussed, but it is clear that the advantages of various values and distributions of uncertainty over the range of the tests must be weighed against the concomitant testing expense associated with possible test designs.

Technique

No estimation of the uncertainty in test results is possible until a test procedure (or design) has been hypothesized; hence we must look first of all at the way test procedures are evolved aside from uncertainty considerations. In the case of performance testing of power machinery, basic procedures are a part of the professional lore and are in part formalized in the power test codes. In the absence of such specific guidance there is a general formula for establishing the essential features of a test:

Define Performance Parameters in Mechanical and Thermodynamic Terms, and classify them according to their independent or dependent character. (This must be partly arbitrary, but the independents usually may be identified by their close relationship to the imposed operating conditions.)

Work Backward From These Semiabstract Parameters to Observable Physical Quantities. Thus the efficiency of a pump is broken down as follows

$$\eta = \frac{\text{Output}}{\text{Input}} = \frac{Q\left[\frac{\text{ft}^3}{s}\right]\Delta P\left[\frac{\text{lb}}{\text{ft}^2}\right]}{T\ [\text{ft lb}]\ \omega\left[\frac{\text{rad}}{s}\right]}$$

The components thus defined are subdivided further in terms of their relationships to actual observable quantities, as follows:

Components	Observable quantities	Relationship
Q = volume rate of flow	ΔH = orifice drop	$Q = K\sqrt{\Delta H}$
ΔP = pressure rise	P_2 = discharge pressure	$\Delta P = P_2 - P_1$
	P_1 = inlet pressure	
T = shaft torque	F_b = net dynamometer load	$T = F_b r_b$
	r_b = dynamometer arm	
ω = shaft angular velocity	n = number of revolutions	$\omega = \frac{2\pi n}{\Delta t}$
	Δt = duration of run	

From the standpoint of the uncertainty analysis it is important

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[2] "Describing Uncertainties in Single-Sample Experiments," by S. J. Kline and F. A. McClintock, *Mechanical Engineering*, vol. 75, 1953, pp. 3–8.

TABLE 1 PORTION OF A TABULAR UNCERTAINTY ANALYSIS

Item i	Upstream pressure	Downstream pressure	Press diff	Sp gr Hg	Metering head ΔH	Orifice const	Flow rate Q
Symbol	P_1	P_2	$P_1 - P_2$		$\frac{P_1 - P_2}{12}(13.6 - 1)$	KA	$KA\sqrt{2g\Delta H}$
Units	in. of Hg under H_2O	in. of Hg under H_2O	in. of Hg under H_2O		ft of H_2O	ft^2	ft^3/s
Typical value, x	10	3	7	13.6	7.35	0.060	1.305
Uncertainty interval w_i (20 to 1)[a]	±0.05 (est.)	±0.05 (est.)	±0.07 (deduced)	±0.01 (est.)	±0.074 (deduced)	±0.0015 (est.)	±0.037 (deduced)

[a] 20 to 1 are the odds offered that the uncertainty interval will include the true value.

that the derivations of these simple quantities from actual instrument readings include corrections for calibrations, conversions, barometric readings, and so on, be indicated.

The tabular method is convenient for this procedure. Each step of the breakdown represents a calculation step which must be carried out in the forward direction when results are being worked up from the data. The step-by-step procedure thus defined is almost always the most efficient method for handling test analyses. The first draft of such a table results from a right-to-left development which starts out like this

	h_2	ΔH		Q	Output	η
etc. ←	in. Hg over H_2O	$(h_1 - h_2)\frac{13.6-1}{12}$	$K\sqrt{\Delta H}$	$Q\Delta P$		$\frac{\text{Output}}{\text{Input}}$
	↑ Observation					

Even this might be elaborated by including a temperature correction for the specific gravity of mercury. This would be considered under the heading of "third-order alternatives" as will be discussed in the next section.

Consider Alternatives. At a certain point in the breakdown of each component entering the final definition of a parameter, a decision must be made with respect to the measurement technique to be used. Decisions made early constitute choices between major alternatives. Selection of an "inferential" method instead of a "direct" measuring process would constitute such a decision. It must be recognized that such quantities as efficiency have alternative definitions

$$\eta = \frac{\text{Output}}{\text{Input}} = \frac{\text{Input} - \text{Losses}}{\text{Input}} = \frac{\text{Output}}{\text{Output} + \text{Losses}}$$

Further down the line relatively minor alternatives appear, e.g., a choice between weigh tanks and headmeters or between torquemeter and cradled dynamometer.

Finally, specific instruments must be chosen. These selections are distinctly third-order (but not necessarily trivial) decisions. There are associated choices such as time and frequency of any particular observation and the degree of refinement to be employed in instrument calibration.

Make Preliminary Uncertainty Estimates. If every alternative test procedure which might be defined by the branching process just described—including all the permutations and combinations of major, minor, and third-order alternatives—had to be subjected to an uncertainty analysis the situation would be hopeless. There are three reasons why such tedious examinations are unnecessary: (*a*) we are not, in general, obligated to find the best test designs; (*b*) we are not devoid of experience and judgment; and (*c*) the manner in which observational uncertainties are combined to qualify the result properly, i.e., as the square root of the sums of the squares of their individually propagated effects, often makes it possible to choose quickly between families of test designs characterized by major alternatives.

If no direct economic advantage or technically important information will accrue from a test of exceptional reliability, any procedure which will keep uncertainties within acceptable limits will fill the bill. Often such a procedure is selected best on the basis of available facilities and instruments, experience of personnel, or individual preference. An obligation remains to verify that the test thus designed will produce acceptable results and to pay some attention to serious imbalances, i.e., concentrations of sources of uncertainty in one or two measurements.

When we consider those cases in which a degree of optimization is sought, we conclude that a mechanism for short-cutting the straightforward analytical procedure must be found. This involves full application of experience to the lumping of uncertainties. It is well known in connection with the making of estimates that if little is known about the subject of the estimation, a detailed breakdown must be made to insure a given reliability in the answer. Conversely, a background of experience will justify a coarser-grained study. The key to the application of this principle is a method of utilizing the backward breakdown for estimating uncertainties to be anticipated in the results. The method is to work forward through the table along a chosen series of alternatives. The uncertainties in individual observations are propagated through the progressive stages of synthesis to the final results. This is illustrated by means of a portion of a hypothetical "front end" of such a breakdown, see Table 1. This portion of the development deals with data relating to a flow measurement. The w's in Table 1 are uncertainty intervals in the corresponding items, estimated or deduced. They are expressed in the units of these items.

Following Kline and McClintock, these intervals are qualified with odds expressing the estimator's confidence that the true values will fall within these limits. These uncertainties are assigned on the basis of judgment to the direct observations and are deduced as follows for the derived quantities

$$w_{(P_1 - P_2)} = \pm\sqrt{\left(\frac{\partial(P_1 - P_2)}{\partial P_1} w_{P_1}\right)^2 + \left(\frac{\partial(P_1 - P_2)}{\partial P_1} w_{P_2}\right)^2}$$

$$= \pm\sqrt{(1 \times 0.05)^2 + (1 \times 0.05)^2} = \pm 0.07 \text{ in. Hg/H}_2O$$

$$w_{\Delta H} = \pm\sqrt{\left(\frac{12.6}{12} \times 0.07\right)^2 + \left(\frac{7}{12} \times 0.01\right)^2} = \pm 0.074 \text{ ft H}_2O$$

$$w_Q = \pm\sqrt{\left(\sqrt{64.4} \times 7.35 \times 0.0015\right)^2 + \left(\frac{1}{2} \times 0.060\sqrt{64.4} \times 0.074\right)^2}$$

$$w_Q = \pm\sqrt{0.00106 + 0.00032}$$

$$= \pm 0.037 \text{ ft}^3/\text{s}$$

$$\frac{w_Q}{Q} = \frac{0.037}{1.305} = \pm 2.8 \text{ per cent}$$

This result is, of course, about what an experienced test engineer would expect from an uncalibrated orifice. It is entirely proper to introduce a directly estimated uncertainty, w_Q, into the evaluation of uncertainty eliminating those steps detailed in the foregoing. Similarly, any uncertainty, w_i, which can be arrived at by divine guidance or through the wisdom of the test designer may be entered directly in the table obviating the necessity for consideration of precedent items. Thus he may evaluate a torquemeter as uncertain to the extent of ±2 per cent (20 to 1) against dynamometer uncertainty of ±1 per cent (20 to 1) and, in view of a requirement for accuracy, discard the torquemeter method without further consideration.

A little experience also helps in taking advantage of propagation by squares. If the principal contributors to the uncertainties associated with major alternatives can be spotted, very good comparisons can be made neglecting the lesser influences. Such partial evaluations of uncertainty are resorted to at some risk of inadvertent omission of a significant source of error. Only sound judgment based on relevant experience can minimize this hazard.

Make Detailed Analysis of Semifinal Design. If patience has not been exhausted, a set of major and minor alternatives now should be selected on the basis of preliminary uncertainty analyses and pertinent economic considerations. Tentative selections of instrumentation then are completed and the corresponding uncertainties in individual observations estimated.

It is beyond the scope of this paper to discuss how this is done, but attention is called to the fact that instrument indications differ from "true" values for many reasons. Not the least of these is lack of correspondence between a sampled quantity and its presumed counterpart in the theoretical model on which the interpretation of the data is based. This may be due to multidimensionality of flow conditions, failure to obtain steady state, and so on. Uncertainties arising in such ways must be estimated and ascribed to the observed values.

These instrument uncertainties then should be propagated step by step to the result. The author favors tabulating the uncertainties in each deduced subquantity as indicated in Table 1.

Complete Test Design by Balancing Uncertainties. A table prepared as described in the preceding paragraph will indicate in which respects the test has been overdesigned and in which respects the design is inadequate. Major contributions to the final uncertainties can be traced quickly to the corresponding observations. Possibilities for reducing the uncertainties then may be weighed—improved calibration to reduce fixed errors; increased numbers of readings to reduce accidental errors; differential instruments instead of differences between independent observations, and so on. No outstandingly offensive determination should remain after this work.

Critical examination of trivial contributors to uncertainties is perhaps less important. No one will recommend deliberate adoption of an inferior instrument in the absence of economic compulsion. However, if superfluous precision has been built into the test design at high cost in terms of equipment or effort, reconsideration is definitely in order.

Take Final Look at Complete Test Procedure. The analyses just outlined will have been concentrated on the operating point of greatest interest. Before attributing the predicted accuracy to a complete series of determinations, analyses should be made near the extremities of the operating range. It is also worth considering how many points are to be run. If the estimated uncertainties are interpreted as fuzziness in the locations of the points and if at least part of such fuzziness is not attributable to fixed errors, then a curve having a character unambiguously defined by these points generally will be less uncertain than the points themselves. The more complicated the curve, the more points are required to insure the absence of ambiguity. For this reason, final decisions as to the number and distribution of determinations often are made as the testing progresses. A healthy suspicion of double inflections is advocated for those charged with the making of these decisions.

Conclusion

A definite procedure for the application of uncertainty analyses to the design of tests has been described. The successful and economic application of this formula depends heavily upon a background of experience and the capacity to distinguish important from trivial decisions. Conscious application of this method will accelerate the development of judgment relative to test procedures.

Discussion

C. B. Haughton, Jr.[3] The writer heartily subscribes to the author's "cookbook procedure" for insuring reliability in power-plant test results, and suggests, in addition to the reasons given by the author, that such a procedure is always justifiable on the following grounds:

1 If the test engineer designs his tests by a step-by-step procedure, his results from test to test, and project to project will achieve a consistency that too often is lacking in long-range as well as in short-range investigations.

2 Many of the cookbook steps are unavoidably routine, but if used as a check list will often save the test engineer's time in setting up new tests. This should not eliminate the engineer's need to think seriously about every step in the procedure, however.

3 If large numbers of test data are to be analyzed later by various statistical procedures, the analyst will have much more confidence in the statistical nature of the data, if he knows the tests were designed consistently according to a carefully prescribed procedure, than if designed by haphazard methods.

Statisticians have been known to wring their hands over an evident lack of control throughout test data which they have been required to analyze, even among data taken from many famous classical tests of the seventeenth, eighteenth, and nineteenth centuries.

4 Testing in many scientific fields is now becoming so complex, and involves many quantities that are so difficult to measure or to anticipate, that a test-design procedure based on tried and true common-sense techniques becomes all the more necessary if maximum utilization of the data is to be obtained.

Thus it all the more behooves research engineers to swallow occasionally their pride of judgment and intuition and adopt test-design procedures that will in the long run benefit them, and will let them apply their best judgment to those phases of the tests which require really sound and original treatment.

The writer wonders why the author did not emphasize a little more the need for a judicious choice of instruments in any test design. In the example used in the paper it is only rather subtly implied that instrument choice should not uncommonly be based on the simple (and sometimes complex) functional relationships among the variables. In the author's equation for flow rate

$$Q = KA\sqrt{2g\Delta H}$$

it can quite easily be shown that the per cent error in the flow rate Q is only one half the per cent error in the measurement of the metering head, ΔH. This is merely another way of saying that if we must measure Q to an accuracy of 1 per cent, we also

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must be prepared to measure ΔH to an accuracy of only 2 per cent. Thus it would be foolish to select a head-measuring instrument having an accuracy of, say, $^1/_2$ per cent. Other considerations, of course, might justify the better instrument but not on the basis of the functional relationship here shown.

The problem, then, of designing a test so that no more than a given uncertainty in the result will occur involves many factors, not the least of which is a good look at the errors in the instruments themselves, as related to the results which they must combine to give.

S. J. Kline.[4] This paper is an exceptionally able discussion of the problems which confront the designer of single-sample experiments. The author's comments show the judgment of a competent and experienced engineer which is typical of his work. The writer has only one comment to make which refers specifically to the material presented in the paper, which is that the author should not apologize for the cookbook approach. Not only is any codified approach an improvement on the method of design so commonly employed in this type of work, that is, ignore the errors in the design stage entirely, but also, it is believed, the approach taken by the author is essentially sound and will pay important dividends to those who employ it. It will pay dividends not only in terms of increased accuracy where required, but also in lower cost for a given total accuracy and in the saving of time and frustrations.

Commenting on the method given in footnote 2 of the paper, which is the underlying method employed by the author, it is necessary to remind ourselves of the meaning of the term single-sample experiment. This is an experiment in which most or all of the uncertainties cannot be found from statistics and must be "guestimated." This is roughly equivalent to saying that it is impossible or at least unfeasible to provide adequate replication. Such is usually the case in the test of power plants owing to the very great cost of each test. This does not mean, however, that we should not replicate where possible; from the point of view of accuracy, replication is always desirable. Nor does it mean that it is profitable to cry over spilt milk by suggesting that all experiments in which replication is not employed are worthless. Certainly such single-sample experiments are not as reliable but they are far from worthless if performed properly.

All of the writer's experience with experiments over the past two and a half years, in which he has employed the method of footnote 2 of the paper, indicates that proper performance of single-sample experiments is dependent primarily on (*a*) analysis of the uncertainties in a thorough fashion as a part of the design stage, (*b*) provision for adequate cross checks in the experimental work including such things as energy balances, known theoretical relations, and so on. It is evident that both of these relate primarily to the design stage. Furthermore, many experiments which ultimately will be multiple-sample are single-sample in the design stage since no data are yet available. This further increases the importance of the technique suggested by the author.

At this point the objection might be raised that if adequate analysis of the uncertainties is made in the design stage, cross checks are unnecessary. This is decidedly untrue. Not only do cross checks provide means for checking the adequacy of the analysis of the uncertainties made in the design stage and for eliminating gross errors, but also they provide a means for improving judgment for future analysis of uncertainties and give the strongest possible type of verification of the reliability of the results. It is this last point which is most important. It must be borne in mind constantly that an analysis of uncertainties based on guestimates for a single-sample experiment can be of great value, but it is still only making the best of an inherently bad situation regarding accuracy.

The need for analysis of uncertainty in the design stage cannot, in the writer's opinion, be overemphasized. This fact is easily seen by considering the relative economies of changing an experimental apparatus in the design stage with that of changing it after the apparatus has been constructed. Certainly not all the pitfalls of experimental work can be avoided in this way but, on the other hand, a large percentage of them can, particularly by an experienced engineer. It has been the writer's experience to date, in those few cases in which he has applied the type of analysis suggested by the author in the design stage, albeit in a less systematic fashion, that the results have been eminently satisfactory. In one case results were obtained, without undue strain, which verified behavior previously predicted theoretically but masked by scatter in all previous experiments.

For these reasons, the writer personally feels that the analysis of uncertainties in the design stage is important enough to justify completely the type of standardized approach suggested by the author. And he firmly believes, as stated previously, that such an approach will pay large dividends in terms of improved accuracy, lower cost, and decreased time and frustration.

Author's Closure

The author wishes to thank Mr. Haughton and Professor Kline for their complimentary remarks and for their vigorous defense of the cookbook procedure. Mr. Haughton's point that the uniform application of such procedure will promote consistency in test results is very well taken indeed. It is somewhat related to Professor Kline's advocacy of the maximum use of cross checks to develop judgment in uncertainty analyses. Natural reluctance to undertake these laborious analyses will be overcome only if such demonstrations of their utility are frequently made.

We are indebted to Professor Kline for his clear exposition of the character of the single-sample experiment and his reiteration of the propriety and practicability of guestimating its uncertainties. It is surely important for engineers to recognize that their experiments inevitably fall into this category; and if forecasts of the reliability of their results are to be made, the methods proposed by Professors Kline and McClintock must be applied pretty much as proposed in this paper.

The author apologizes for the indirectness of the paper's attack on the problem of instrumentation selection and for failure to stress the virtues of repeated observations where they are possible. There are many opportunities for papers which will deal adequately with such topics. It is to be hoped that these papers will be forthcoming.

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[4] Assistant Professor of Mechanical Engineering, Stanford University, Stanford, Calif. Mem. ASME.

Measurement Errors—Classification and Interpretation

By J. C. BOONSHAFT,[1] HATBORO, PA.

The classification and appearance of errors contributing to the uncertainty of measurement with industrial indicating and recording instruments are described, together with design details and measurement situations which affect them. Relationships and typical figures are given to show technique of evaluation of the precision, or internal consistency, of these instruments.

Introduction

RELIABLE measurement is based on a complete understanding of the appearance of errors, and how they are affected by the details of construction and method of use of the measuring instrument.

If we consider a single instrument which measures any physical quantity or variable and translates its measurement into useful intelligence as a scale indication by whatever means this may be achieved, the accuracy of this instrument is the maximum expected difference between its indications and the true value of the variable which it measures. This usually is described in terms of either ± per cent of full scale range at any point of the scale, or ± per cent of instantaneous reading. Either method is acceptable provided the range is given. Accuracy involves a true value and is a composite of the following parts: (1) How well is the true value known? (2) How well does the instrument measure this true value? (3) How well does the experimental method reduce uncertainties in comparison?

The true value of a variable usually is represented by fixed points or by standards calibrated at these fixed points with scales interpolated by defined methods. Useful realization of these fixed points generally is impractical industrially, and the true value in instrument calibrations is represented by corrected indications of secondary standards.

The "true" value may then be only an approximation. Certainly the accuracy of an instrument must include this uncertainty. However, this uncertainty usually can be reduced by selection of better standards, or superior intercomparison methods. What ultimate care in method and selection of standards is warranted by an instrument can be determined only by knowing with what uncertainty the instrument itself being calibrated could measure a variable if the value of that variable has no error. This is its precision, or maximum variation in indication that will occur in repeated measurements of the same true value of the variable. It is, therefore, a measure of the internal consistency or repeatability of measurement with an instrument. It is expressed as the ± difference between any measurement and the average indication for a given true value found from several independent determinations.

In this paper we will limit our subject to recognizing the factors of which a numerical value of precision is composed in industrial measuring instruments of the indicating and recording type. Because of the popularity of the pneumatic transmitting class of instrument, principal consideration will be mechanical-pneumatic transmission and receiver devices.

The measurements from which a numerical value of precision for such instruments is determined must include a sufficient number to recognize the appearance of errors because of the following: (1) scale factors, (2) direction of approach of variable to final reading, (3) just past history of the measurement system, (4) length of time at reading, (5) life history of the measurement system, and (6) ambient factors, particularly temperature.

Scale Factors

The ± variation in reading of a measurement for a given fixed scale indication gives the useful part of the sensitivity of an instrument, or "readability." It should be recognized that the best readability of a single indication regardless of sensitivity is only 0.1 to 0.2 of the smallest scale division,[2] unless aids such as verniers are employed. Such acceptable values of readability may be degraded rapidly by a poor physical layout of the scale or chart. Bold division lines and pointers lead to good recognition from a distance, but make for poor readability because of the need to consider the center of these division lines as a reference when interpolating. The distance between scale and pointer may be important when instruments are mounted at other than eye level, for large separations lead to errors of parallax, or differences in reading based on direction of observation with changing position of observer.

Direction of Approach

If the variable moves slowly or in very small steps away from a given value, a certain change of variable occurs before the instrument responds by change in indication. This is its sensitiveness, a value of which can be found with a variable changing in either direction. The span including both values of sensitiveness is the dead zone. The dead zone also can be demonstrated if the variable approaches but does not overshoot a given true value from

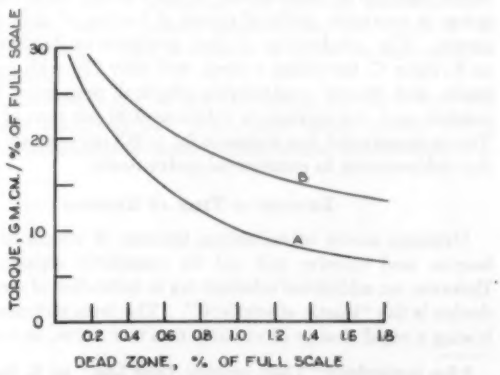

Fig. 1 Element Torque and Dead Zone

[1] Chief Engineer, Fischer & Porter Company.

Contributed by the Power Test Codes Committee and presented at the Annual Meeting, New York, N. Y., November 29–December 4, 1953, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, October 7, 1953. Paper No. 53—A-219.

[2] "Tenths Estimating in Scale Reading," by H. Backstrom, *Zeitschrift für Instrumentenkunde*, vol. 50, 1930, pp. 561–575, 609–624, 665–679.

both up and down scale directions. The difference between the final indications is the dead zone.

The sensitiveness of an instrument is related almost entirely to the force/friction ratio of the receiving or measuring element and the linkage system and accessory units which the element is required to drive.

In a simple recorder with scale length of 4 to 5 in., curve A in Fig. 1 shows the relationship of dead zone to element power. This may vary slightly with different makes of instrument. Torques shown are measured at the pen spindle.

Each additional accessory in the instrument adds a frictional load which increases the required element torque for a given dead zone. Curve B in Fig. 1 shows this in an indicating-recording-integrating-controlling instrument. A field approximation of element torque can be made easily by suspending a weight on the pen (with static variable) measuring excursion and effective pen arm from its pivot point, where

$$\text{Torque at pen spindle (gm cm/per cent of scale)} = \frac{\text{Effective arm, cm} \times \text{load, gm}}{\text{Excursion in per cent of scale}}$$

This simple technique may be used to determine if the dead zone is reasonable for a given instrument, or if such service problems exist as tight or dirty bearings, rubbing links, or high pen load on chart. A suggested loading weight is 5 grams, easily made from coiled solder. Measurements should be made with initial indication in the upper part of the scale, where the effective arm is largest and more accurately measured.

Just Past History

Sensitiveness usually is measured with only small excursions of the variable, and generally is a fixed value throughout the range with a linear element. A second direction factor, hysteresis, is measured throughout the range and will vary in magnitude depending on just past history of the measurement system. If a full-cycle calibration of an instrument is made, the variable, for example, being manipulated through the instrument range from 0 to 100 per cent and then back to 0 per cent, hysteresis is the lagging behind of indication during the back calibration. By definition, hysteresis is zero at 100 per cent and 0 per cent of scale, and in a mechanical instrument is a maximum usually at 60 to 70 per cent. Hysteresis can be demonstrated through a full-cycle calibration covering only part of the range. Hysteresis in this calibration will be zero at the limits of the calibration, and greatest at 60 to 70 per cent of the span covered. Its magnitude as compared with full-scale calibration hysteresis may be approximately proportional to the fraction of the span covered.

In a mechanical instrument, hysteresis is the result of inelastic behavior of a spring, bourdon, bellows, and the like, because of either material of construction, or such design factors as working stress or reversible shifts of points of loading of such spring elements. The availability of new precipitation-hardening alloys as Ni Span C, beryllium copper, and alloy 720 with good elastic limits and closely controllable physical properties has made possible such low hysteresis values as 0.03 per cent of full scale. This is exceptional, but values of 0.1 to 0.3 per cent are a present-day achievement in commercial instruments.

Length of Time at Reading

Dynamic errors in indication because of effects of capacity, inertia, and transfer will not be considered separately here.[3] However, an additional inherent lag in indication of a mechanical device is the "elastic aftereffect." The term indicates that following a rapid change of variable to a new value, or return to its original value, the instrument indication may not stabilize immediately but will be delayed. The delay when short is not recognizable and is considered part of the usual lag coefficient. With certain materials, principally inexpensive steels used in many instrument springs, instantaneous errors as high as $^1/_2$ per cent of full scale may be demonstrated, with full recovery to no error delayed as long as 24 hr. This effect becomes important only in high-precision work with rapidly changing variables.

Life History

Depending upon the application of a given instrument, time or use factors may become important. The time effect of permanent set, or change in calibration because of creep or plastic flow at high stress within the measuring element, is an error easily eliminated by periodic recalibration. Other time effects, as, for example, in a simple bourdon temperature element, may be the result of chemical changes in the filling fluid, or slow leakage of gas filling.

Permanent set can be seen as a uniform shift of indication throughout the scale known as zero shift, or a range shift proportional to reading, or combinations of both. It therefore requires calibrations at two points or more of the scale to minimize these errors.

The usual industrial instrument is designed and constructed for long life, where use itself will not cause error. While difficult to extract use factors from time factors in instrument error, the service in which an instrument is applied may be used to set the frequency of recalibration. Experimentally, the change in calibration with use of a typical pressure element is shown in Fig. 2.

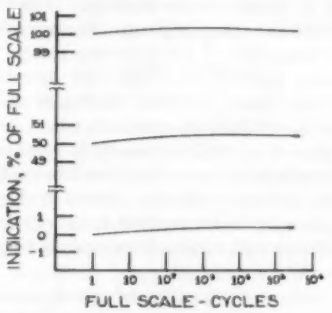

Fig. 2 Life History of Pneumatic Receiver

The changes of calibration shown occur with full-scale pulsing at a rate of 500 per hr. Complete data are not available to determine if these changes are additionally a function of pulse rate.

The figure demonstrates the advisability of "breaking-in" such elements by full-scale cycling before putting into use. This generally is done by the instrument manufacturer, as a test designed to find poor construction. The figure also shows that it is advisable to follow closely the element in the field by calibration check at the first 1000 cycles, with repeat checks at possibly 100,000-cycle intervals thereafter.

Ambient Factors

The principal ambient factor in mechanical instruments is temperature. Ambient pressure may be of limited importance in certain types of elements, particularly the "force-balance" pneumatic transmitter.

Ambient-temperature errors may be of the zero-shift or range-shift types, or in combination. The principal error is that of the change of elastic modulus and, consequently, range calibration, of bourdons, springs, and bellows. Concomitant zero shifts may occur, as well as zero and range shifts, because of expansion of

[3] See particularly "Thermometric Time Lag," by R. Beck, Trans. ASME, vol. 63, 1941, pp. 531–538.

linkage parts. With usual materials, the expected range shift because of change of modulus is 1 per cent per 50 deg F change in ambient, but this may be modified by other effects. Certain zero thermoelastic coefficient materials as Ni Span C and isoelastic alloy exhibit a much lower and usually negligible range shift. Invar is used frequently to reduce zero errors based on expansion of link parts.

Fig. 3 shows the effect of change in ambient temperature for two motion-type pneumatic transmitters that are correct at 80 F and demonstrates the importance of knowing construction details in estimating ambient-temperature errors.

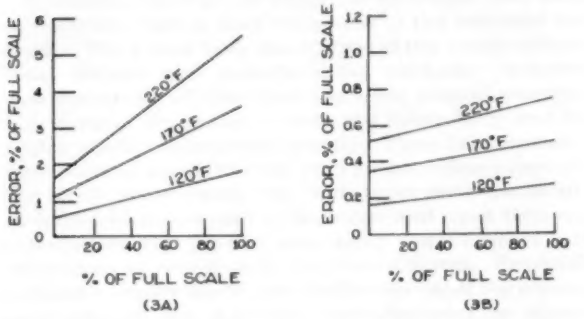

FIG. 3 PNEUMATIC-TRANSMITTER AMBIENT-TEMPERATURE EFFECTS

Fig. 3(A) is a transmitter constructed of usual materials, i.e., brass bellows, 18–8 stainless links, steel calibrating spring. Fig. 3(*b*) is a transmitter designed particularly to reduce these errors. Refinements such as zero thermoelastic springs and zero-expansion links are responsible for the superior performance shown.

It is possible, but difficult, to prophesy the exact ambient-temperature effect in an instrument. A suggested procedure is that of calibrating the instrument through temperature extremes in an oven or, if impossible, checking calibration twice in a single day at a season where temperature fluctuates rapidly. Fig. 3 shows the need of such calibration at only 2 or 3 points of the scale, since effects are quite proportional to indication.

A secondary temperature effect of importance only at high ambients is the acceleration of permanent set as the elastic limit of calibrating springs reduces. This again is a function of material of construction and differs widely among accepted spring materials. Suffice to caution that the interval between periodic recalibrations be reduced for instruments operating at ambient temperatures above 150 F.

Ambient-temperature effects in other types of instruments vary widely. A comparison of three constructions of industrial mercury manometers is shown in Fig. 4. Curve A shows a simple manometer of steel, curve B is a manometer using a temperature compensator of one type, and curve C shows effects with a superior type of temperature compensator. All curves are based on an ambient-temperature change from 70 to 170 F, for 100-in. water instruments. The importance of knowing construction details in estimating measurement errors is indicated by this figure.

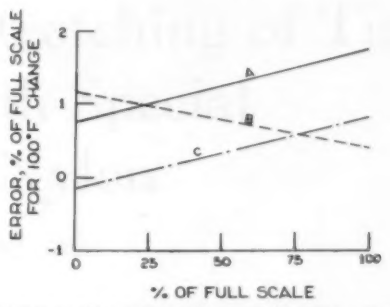

FIG. 4 MERCURY-MANOMETER AMBIENT-TEMPERATURE EFFECTS

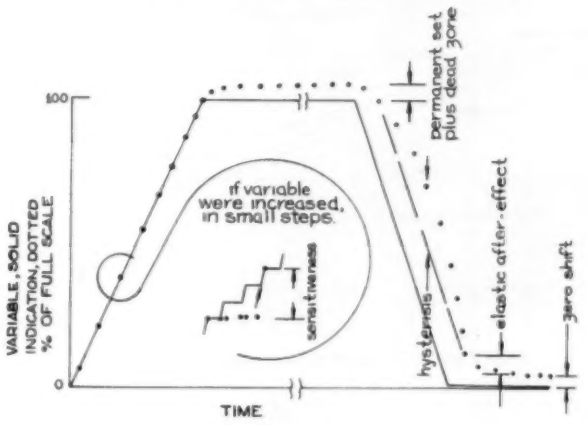

FIG. 5 COMBINED ERRORS

SUMMARY

Fig. 5 shows the usefulness of this information in extracting sources of error, which then permits evaluation of the measuring instrument for a given task. Dynamic errors are not considered in this figure, which is based on a slowly changing variable.

The relationship of measurement uncertainties, in usual order of magnitude from the smallest to the largest, is as follows: ± readability, ± sensitiveness, dead zone, elastic aftereffect, hysteresis, ± permanent set, ± precision.

Readability, depending on sensitivity, may be equal to or larger than sensitiveness. This is because sensitiveness is a measure determined from relative indications while readability is an absolute value. It is possible to discern a change in indication more closely than that change can be read.

Dead zone is equal to plus and minus sensitiveness. Precision is usually equal to sensitiveness plus readability, plus hysteresis/2, where permanent set is minimized by periodic recalibrations. In the foregoing listing, certain uncertainties are shown as ± while others are not. This is a matter of definition and must be followed accurately.

With an understanding of the appearance of errors and the measurement situations and instrument design details which affect them, techniques to reduce measurement uncertainties can be applied.

The Latexing and Hot-Stretching of Tire-Cord Fabrics With Special Emphasis on Nylon

By C. A. LITZLER,[1] CLEVELAND, OHIO

It is estimated that 1.67 billions of tires have been made in the United States since the advent of the motorcar and truck. There have been many types of tire constructions, tread designs, and manufacturing methods; however, now practically all tires have the same general construction design. Processing of both the fabric body and the rubber stocks has improved greatly. These improvements have resulted in a better tire with longer mileage capacity and much better service life. This paper deals specifically with cord fabrics as used in passenger and truck tires and summarizes the general processing requirements and techniques as carried on in the United States. Emphasis is placed on nylon fabric, principally because of the present tendencies of the American manufacturers to accept nylon-cord fabric as being superior to cotton and rayon for many important tire-cord requirements.

Cotton-Cord Fabric

COTTON-CORD fabric has been used in the manufacture of hundreds of millions of pneumatic tires. Most cotton tire cords are of 12/4/2 construction and measure about 0.032 in. diam. The warp cords are constructed of $1^1/_{16}$ to $1^1/_{32}$-in-long fibers spun from 17 to 20 twists per in. and then cabled through approximately $9^1/_2$ to $10^1/_2$ twists per in.

The breaking strength in the treated individual cord varies in the neighborhood of from 15 to 20 lb. The cord fabric is woven with two to three pick threads per in., while the number of ends in a web fabric varies from 23 to 34 ends per in. The mill widths of the fabric vary from 56 to 66 in. and the roll lengths usually are 500 to 1000 yd, depending upon specifications. Characterizations of 12/4/2 cotton are shown in Fig. 1.

Cotton, as the major material in the pneumatic-tire construction, has been superseded almost completely by rayon. Some cotton is still used for tire construction outside the United States, which cotton is usually cemented or gum-dipped prior to calendering.

Rayon Cord

Superseding and replacing cotton cord to form the first all-synthetic carcass is the high-tenacity rayon cord. It is furnished in woven fabric form or on spools for creel operation. The sizes of the basic cords are 1100 denier 2 ply, Fig. 2; 1650 denier 2 ply; and 2200 denier 2 ply. The most common is the 1650/2 fabric, Fig. 3, which is used for most passenger-car, truck, and heavy-duty tires. The fabric is woven 2 to 3 picks per in. with warp cord end count ranging from 11 to 36 ends per in. of fabric width depending upon the ply stock desired. The fabric is furnished by the textile mill in 1000-yd rolls and in widths of from 57 to 62 in. The end count varies with the ply use of the particular fabric.

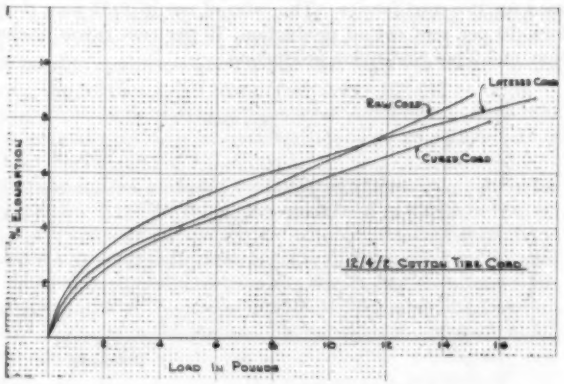

Fig. 1 Characteristics of 12/4/2 Cotton Fabric

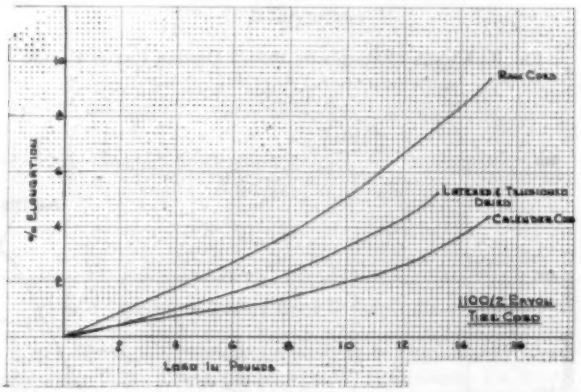

Fig. 2 Characteristics of 1100/2 Denier Rayon Fabric

Nylon Tire Cord

In recent years, heavy-duty truck and aircraft tires have been constructed of nylon-cord fabric. When this cord material is treated properly, it has a high tensile strength and low elongation at work loading. Its most important advantage is its particular quality of resisting road shock. This is due to the high impact absorption of prestretched nylon cord. Its heat and moisture resistance are considerably superior to rayon.

Practically all nylon-cord fabric in the United States is made from Type 300, bright, high-tenacity nylon yarn—two deniers, 840–140 and 210–34. In the 840-140 denier a single yarn is twisted 12 to 15 turns and is plied or cabled and twisted from 10

[1] President, C. A. Litzler Company, Inc.

Contributed by the Rubber and Plastics Division and presented at the Annual Meeting, New York, N. Y., November 29–December 4, 1953, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, November 6, 1953. This paper was not preprinted.

to 15 additional turns. In 210 denier, four single yarns are doubled and twisted to form the 840 denier. Both yarns are practically the same and the reference is based upon the manufacturing procedure. Cord diameters are about 0.021 in. Woven-fabric widths vary from 48 to 60 in. and the end count from 18 to 39 ends per in.

Cotton-cord fabric has been almost completely replaced by rayon. Rayon is being replaced, at a very steady rate, by nylon for many types of tires.

Recent statistics, Fig. 4, show that 92,000,000 lb of cotton fabric were produced in 1952 for tire construction. In the same period 384,000,000 lb of rayon and 8,000,000 lb of nylon were used for tire-cord fabrics. The ratio of use in 1952 was, therefore, 19 per cent cotton production, 79 per cent rayon production, and 2 per cent nylon production.

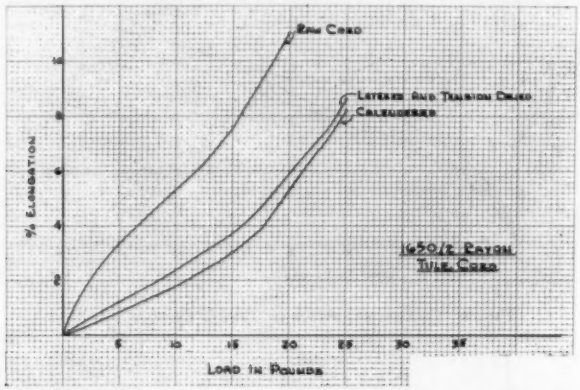

Fig. 3 Characteristics of 1650/2 Denier Rayon Fabric

First-quarter trends of 1953 indicated productions of 93.6 per cent rayon, 3.2 per cent cotton, and 3.8 per cent nylon tire-cord fabric.

These statistics are not in themselves indicative of any real trend. However, increased planned nylon-producing capacity and nylon-latexing equipment definitely show a very heavy trend toward nylon and its general acceptance by the rubber-tire industry and consumer public.

The current production of nylon, suitable for cord fabric, is at the current rate of 16,000,000 lb per year. At this time, plant capacity is being built to provide a nylon production of 45,000,000 lb at the end of 1954.

This increase in nylon production has been brought about by the popular acceptance of premium tires by American motorists. American tire companies are now installing a relatively large number of nylon latexing and hot-stretching machines. These machines will be in operation in the United States by the end of 1953 and 1954, and they will have the production capacity to hot-stretch all of the nylon tire cord produced.

One major tire producer already has had nylon tires in passenger-car sizes on the market for the past year in appreciable quantity. There are approximately seven other rubber companies who have very recently or are currently being equipped with facilities for the proper hot-stretching of nylon cord. These units will produce a stable all-nylon passenger tire carcass cord fabric material.

The general requirement for hot-stretched nylon in passenger-car tires is shown in a family of curves shown in Fig. 5. This series of curves indicates the extensibility of raw nylon, nylon hot-stretched at 1 lb per cord tension, nylon hot-stretched at 7 lb per cord tension, and nylon tension at 5.5 lb per cord.

The elongation of the cord sample marked A indicates a stretch of approximately 3 per cent at 5-lb load and a 5.75 per cent

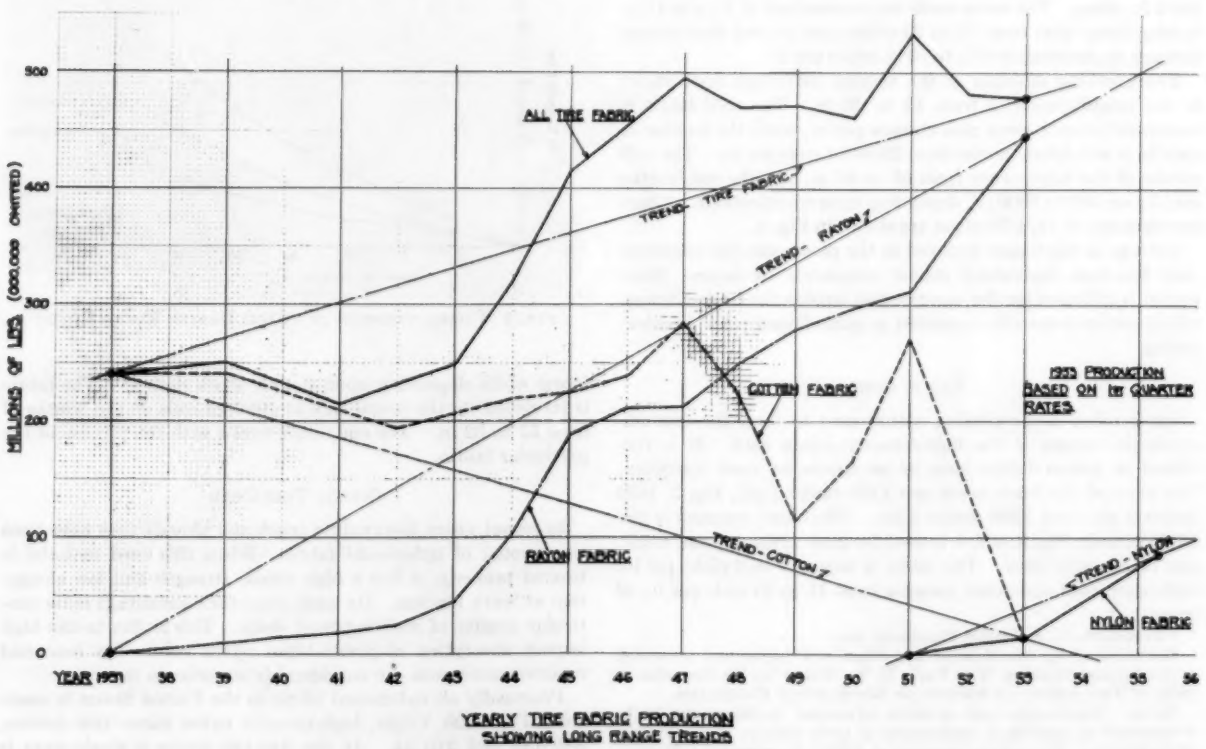

Fig. 4 Yearly Tire-Fabric Production

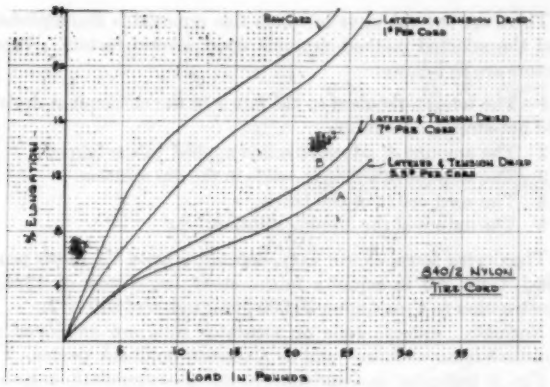
Raw Cord
Latexed & Tension Dried 1% Per Cord
Latexed & Tension Dried 7% Per Cord
Latexed & Tension Dried 3.5% Per Cord
840/2 Nylon Tire Cord
% Elongation
Load In Pounds

FIG. 5 REQUIREMENTS FOR HOT-STRETCHED NYLON-CORD TIRES

stretch at the 10-lb load. The tensile strength is approximately 27.25 lb and the elongation at break is approximately 14 per cent. Although cord specifications vary somewhat with different tire manufacturers, a nylon-cord fabric, stabilized to within this general elongation versus load condition, is quite acceptable for tire construction.

Referring again to Fig. 4, the process of simultaneous latex application, tension-drying, and hot-stretching has decreased the elongation of the cord and increased its tensile strength. The latexing adhesive dip has provided a bond for the calendered rubber and the hot-stretching has stabilized the cord fabric as to its elongation under load. Accordingly, this heat-treating operation removes the possibility of carcass growth in the completed tire.

The hot-stretching and latexing operation is an important step and one which develops optimum strength in the cord for maximum service in the tire.

Fundamentally, the hot-stretching operation requires the immersion of the raw cord, under low cord tension, into water-solution latex or other adhesives. After impregnation, the free excess moisture must be removed mechanically and the fabric subjected to sufficient heated air to remove the final dipping moisture. Simultaneously, the same drying air provides thermal energy to reorganize the molecular structure of the nylon under accurate tension and elongation conditions to effect the proper hot-stretching of the cords.

Hot-stretching, as it applies to nylon, is simply a tension-drying operation that employs considerably higher temperatures and considerably greater amounts of mechanical stretching energy than most other cord materials to stabilize the cord and to prevent additional cord stretch in the completed tire.

The operation consists basically of the impregnation and the tension-drying of the cord under conditions of accurately controlled temperature, heating-exposure times, and stretching tension while the cord is passing through the impregnation and hot-stretching equipment. Time, temperature, and tension are the three variables in the process which must be co-ordinated precisely to produce a stabilized nylon-cord tire fabric.

Rayon, cotton, and nylon all require a similar stretching or tension-drying operation to develop proper cord condition, but cotton and rayon require a considerably lesser amount of tension and temperature than nylon but considerably greater amounts of heating time.

THE DIPPING OR SATURATING OPERATION

Because nylon tire cord has a relatively smooth surface, special latex dips or adhesives are required as surface treatment for the bonding of the calendered rubber.

Some processes employ a double dipping solution. The first dip material is an amide material and the second dip is a modified latex or rubber solution. Recently, special dips have been developed which have a very high adhesion rate; accordingly, full-width fabric units in operation, under erection, or current design, are provided with only single dip operations.

Much work has been done on vinyl-pyridine latex, and its use as an adhesive is now standard practice with several major companies. This water-dip solution gives cord adhesion in the order of 25 lb. It is applied in a water solution containing approximately 17 per cent resorcinol-formaldehyde resin solution and approximately 83 per cent of pyridine latex. This adhesive mixture, applied to normal nylon cord at a cord-dipping tension of $^1/_4$ lb at a 0.5-sec immersion, properly dried and cured as required, will provide approximately 7.5 per cent latex (as dried) to the cord. Dip-solution solids are approximately 12 to 15 per cent.

Fig. 6 gives information on rubber-to-cord adhesion in pounds with various conditions of pyridine latex and at various curing times.

It had long been held that the immersion time of the cord in the dip solution directly affected the amount of solids picked up by the cord. Recent laboratory work has shown that there is no direct relationship in the amount of solids picked up by the cord in proportion to the immersion times. It has been found, experimentally, that the two most important factors affecting dip-solids pickup are cord tensions during dipping and cord finish. Although the data shown are for rayon cord, preliminary test data on nylon fabric indicate the same general conclusions.

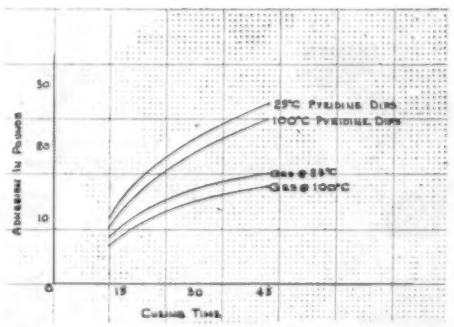

FIG. 6 ADHESIVE QUALITIES OF PYRIDINE-LATEX DIP

Rayon fabric, with various proportions of sulphonated oil or waxy ester finish, has a curve somewhere between the waxy ester and the sulphonated oil as shown in Fig. 7. The waxy ester, because of the lack of absorbency, presents an easier drying problem than the sulphonated-oil finish with its higher solids pickup. No accurate figures or data are yet completed on nylon-cord-fabric solids pickup. It appears from these preliminary figures that nylon cord behaves similarly to waxy ester-finish rayon.

In normal practices, solids in the dipping solutions average 12 per cent. This solid-water ratio is about the cleanest solution to handle in the saturator. Latex build-up on the squeeze rolls and pick-off of the dip solids on subsequent handling rolls or in the drying and hot-stretch unit are at a minimum with low dip solids. With dip solids in the order of 20 to 25 per cent, the use of squeeze or wringing rolls presents serious maintenance problems. Other means, such as burnishing rolls or mechanical moisture-removal devices such as high-pressure air jets, are used. These moisture-removal devices provide some additional maintenance problems and they must be kept properly cleaned. If clean fabric and economic operation are desired, less difficulties are encountered with low solids dip materials than are experienced with solutions of higher solid contents.

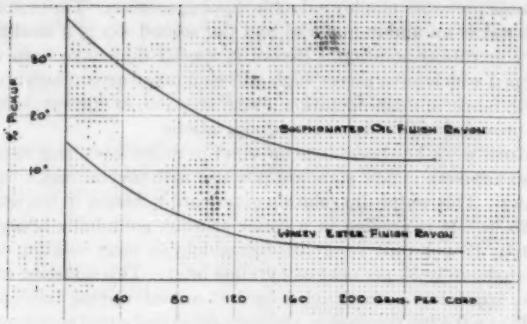

Fig. 7 Cord Tensions Versus Solids Pickups in Dip Tank

The amount of solids applied to the cord extends from 4 per cent by weight, minimum, to a maximum of 10 per cent by weight. The average weight of applied latex in the United States is in the order of 6 to 8 per cent of the dry-weight fabric.

Removal of Excess Dipping Solutions

A critical phase in the cord-saturation operation is the removal of the excess solution and dip solids from the fabric. If the fabric specifications require 6 per cent latex solids on the fabric, this rate must be maintained uniformly regardless of fabric speed or tension.

There are several ways in which this removal is accomplished and the selected method depends upon several factors. These factors are as follows:

1 The tension required in the fabric in the subsequent drying and hot-stretch operation.
2 The amount of adhesive solids applied to the fabric.
3 The amount of adhesive solids in the dip solution.
4 The cord count of the fabric.
5 The nature of the dipping solution.

The drying tensions will vary, depending upon the fabric processed, from 500 lb across a fabric sheet for cotton of 12/4/3 construction to a maximum 20,000 lb for nylon of 840/3/2.

Normally, squeeze rolls or saturator rolls are employed as moisture-removal means and as a tension holdback device.

Fig. 8 indicates various squeeze-roll arrangements, together with a statement of the approximate holdback capacity of the various roll-stand arrangements. The effects of squeeze pressure at the roll nip, the area of fabric wrap, and the friction coefficient of the wet fabric on steel and/or rubber roll faces directly determine the number of rolls required.

The pressure at the nips of any two contacting rolls will vary from 5000 lb across the roll face to a maximum of approximately 25,000 lb for the removal of excess moisture.

Lower nip pressures are desirable in order to prevent crushing or bruising the fabric in its passage through the roll nip. Excessive nip pressures tend to weaken both the warp and pick cords

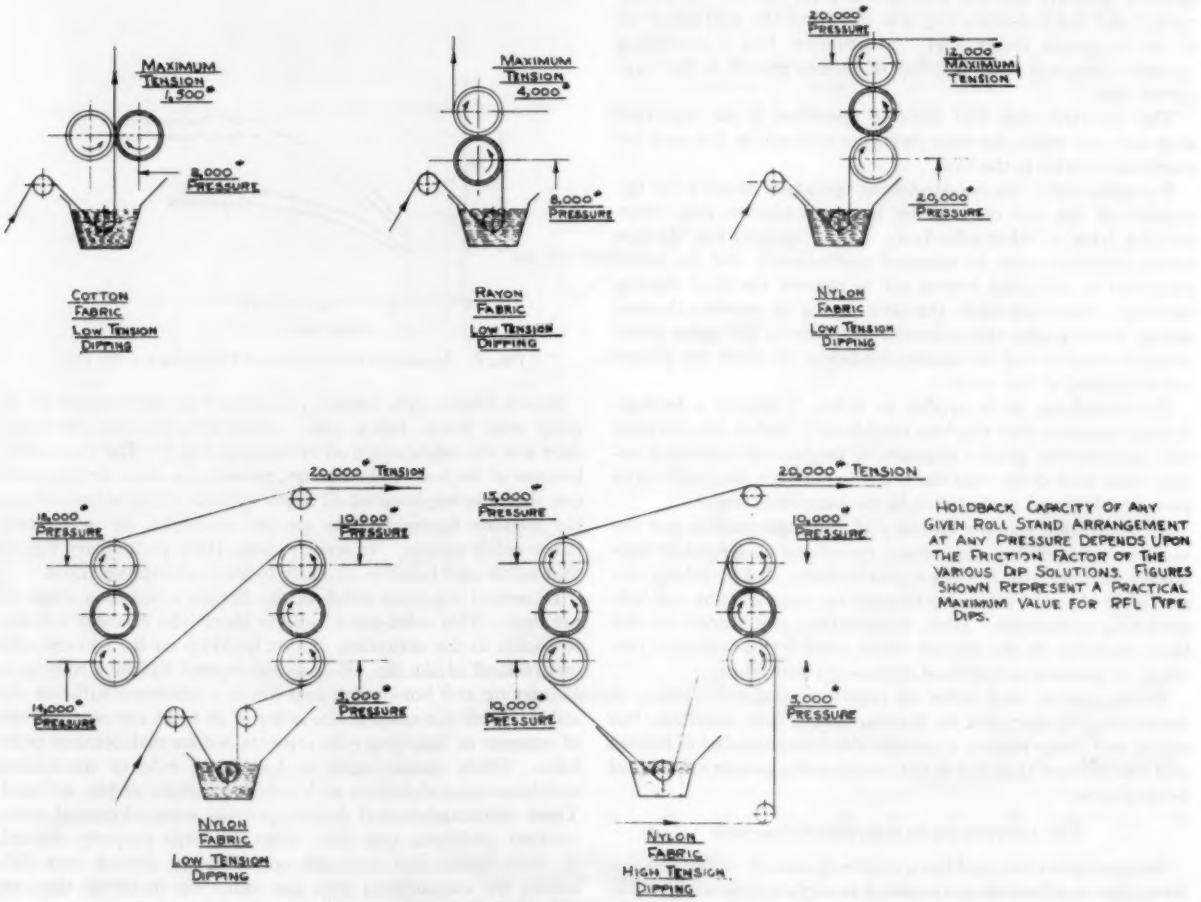
Maximum Tension 4,500
8,000 Pressure
Cotton Fabric Low Tension Dipping
Maximum Tension 4,000
8,000 Pressure
Rayon Fabric Low Tension Dipping
20,000 Pressure
12,000 Maximum Tension
20,000 Pressure
Nylon Fabric Low Tension Dipping
20,000 Tension
18,000 Pressure
10,000 Pressure
14,000 Pressure
3,000 Pressure
Nylon Fabric Low Tension Dipping
20,000 Tension
15,000 Pressure
10,000 Pressure
10,000 Pressure
3,000 Pressure
Nylon Fabric High Tension Dipping
Holdback Capacity of Any Given Roll Stand Arrangement at Any Pressure Depends Upon the Friction Factor of the Various Dip Solutions. Figures Shown Represent a Practical Maximum Value for RFL Type Dips.

Fig. 8 Squeeze and Holdback-Roll Arrangements; High-Pressure Nips

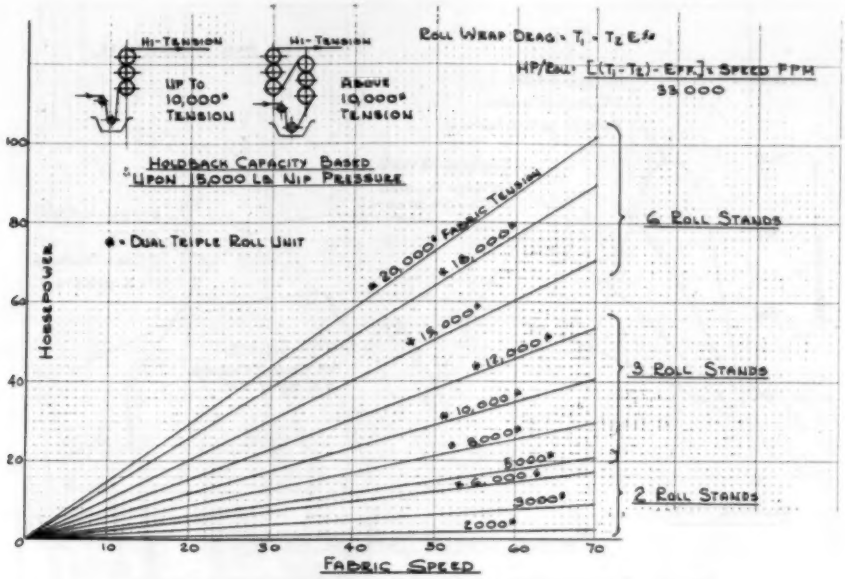

FIG. 9 HORSEPOWER DATA FOR SATURATOR SQUEEZE ROLLS

because of the abrasive action at the point where the warp and pick cords cross. Holdback and pull roll stands with no nip pressure are now being built.

The actual energy to hold back the fabric to develop the required tension is provided by d-c operated squeeze-roll drive holdback motors. This holdback-drive motor, which is one of two in the hot-stretch system proper, acts as a feedback generator into the main electric generator of the system drive. The selection of motor size is critical. The holdback motor size is determined by the operating speed, the holdback tension required, and the nip pressure.

Fig. 9 shows a family of curves giving horsepower relationship to speed and tensions of the saturator squeeze and holdback-roll stand.

Fig. 10 indicates the drive horsepower relationship of speed and tension of the dry end-pull rolls.

ACTUAL HOT-STRETCHING OPERATION

After solution-dipping and the removal of excess dipping-solution moisture, the fabric thereupon enters the hot-stretch unit proper. The hot-stretching operation is normally conducted at temperatures and drying and stretching cycles as indicated in Fig. 11.

The hot-stretch operation is essentially an elongation of the cord fabric under conditions of constant temperature, constant exposure time, and constant tension. The fabric tension must be maintained within suitable holdback rolls and forward-pull rolls as external tension is applied to the cord by appropriate mechanical or hydraulic means. This entire operation is conducted as shown diagrammatically in Fig. 12.

The saturator-roll stand and the main pull-roll stand obviously must be designed to allow forward propulsion and holdback of the fabric without slippage between the rolls because a relatively constant length of fabric must be maintained continuously between these two roll stands. In a typical operation these rolls must be of sufficient mechanical stretch to hold back and subsequently to tension a full-width, 2000-end cord fabric at a maximum tension of 10 lb per cord or total tensions of 20,000 lb.

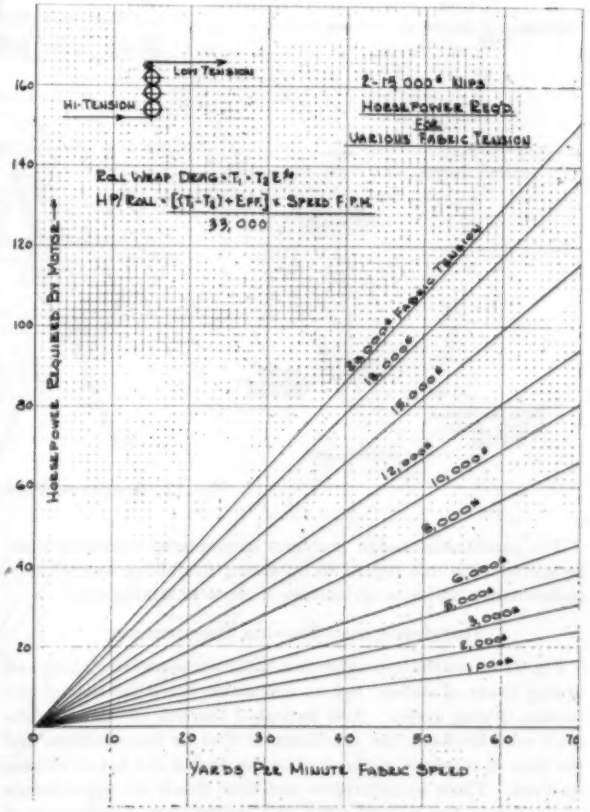

FIG. 10 THREE ROLL-PULL ROLLS—DRY FABRIC

After determining the design of a suitable roll system and a tensioning system of adequate size and strength, the heating and cooling requirements are next in order of importance.

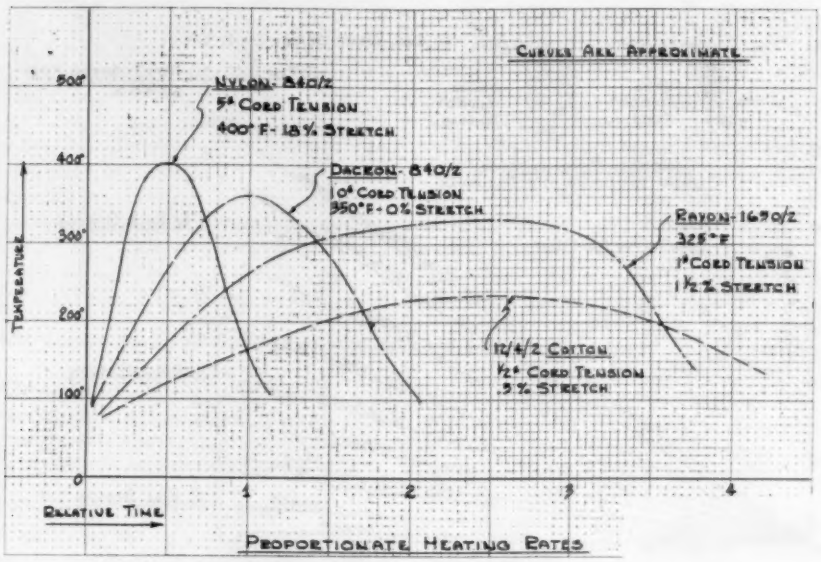

FIG. 11 TEMPERATURES AND DRYING AND HOT-STRETCHING CYCLES

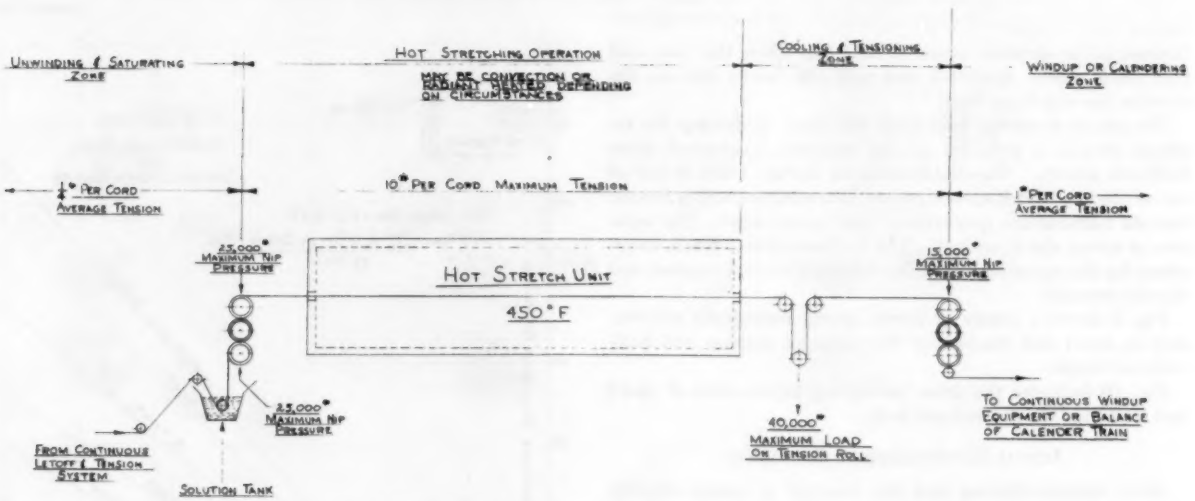
UNWINDING & SATURATING ZONE
HOT STRETCHING OPERATION
MAY BE CONVECTION OR RADIANT HEATED DEPENDING ON CIRCUMSTANCES
COOLING & TENSIONING ZONE
WINDUP OR CALENDERING ZONE
½# PER CORD AVERAGE TENSION
10# PER CORD MAXIMUM TENSION
1# PER CORD AVERAGE TENSION
25,000# MAXIMUM NIP PRESSURE
HOT STRETCH UNIT
450°F
15,000# MAXIMUM NIP PRESSURE
25,000# MAXIMUM NIP PRESSURE
40,000# MAXIMUM LOAD ON TENSION ROLL
TO CONTINUOUS WINDUP EQUIPMENT OR BALANCE OF CALENDER TRAIN
FROM CONTINUOUS LETOFF & TENSION SYSTEM
SOLUTION TANK

FIG. 12 SCHEMATIC VIEW OF HOT-STRETCH OPERATION

The mechanical design problems encountered obviously must be solved with due regard to operating flexibility, accessibility, maintenance, and appropriateness of physical arrangement.

HOT-STRETCHING-TENSION REQUIREMENT

Fig. 11 shows the approximate temperatures and stretching and drying times of cotton, rayon, and nylon cords throughout the tension drying cycles. Also indicated thereon are the approximate cord tensions, the maximum critical air temperatures, and the time of exposure of the cord to the drying and hot-stretching medium. These temperatures and time charts are approximate and indicate the general time and temperature relationship of various cords to each other. These curves have been based upon empirical experimental data provided by measurements of cord weights, taken at various stages throughout the treating cycle and plotted against a relative drying time.

The amount of the total heat absorbed by the nylon cord itself is the most critical factor in the hot-stretching operation. The maintenance of proper air temperature and impact velocity on the cord are of primary importance. Electronic instrumentation with a high speed of response is required for accurate temperature control so that varying fabric speeds will not affect the temperature in the heating medium. It has been found that convection air temperature, air-impact velocity, and exposure times must be kept constant, regardless of fabric speed or mass, to insure a uniform application of thermal energy to the material in the hot-stretch operation.

The intrinsic work that must be performed in the hot-stretching of nylon is the product of the mechanical energy applied in the form of tension or stretch during the heat-treatment operation and the thermal energy applied to and absorbed by the nylon itself. Thermal energy softens the cord and thereby increases its ductility. As the ductility increases, the tension or force required to produce a given stretch is relatively reduced.

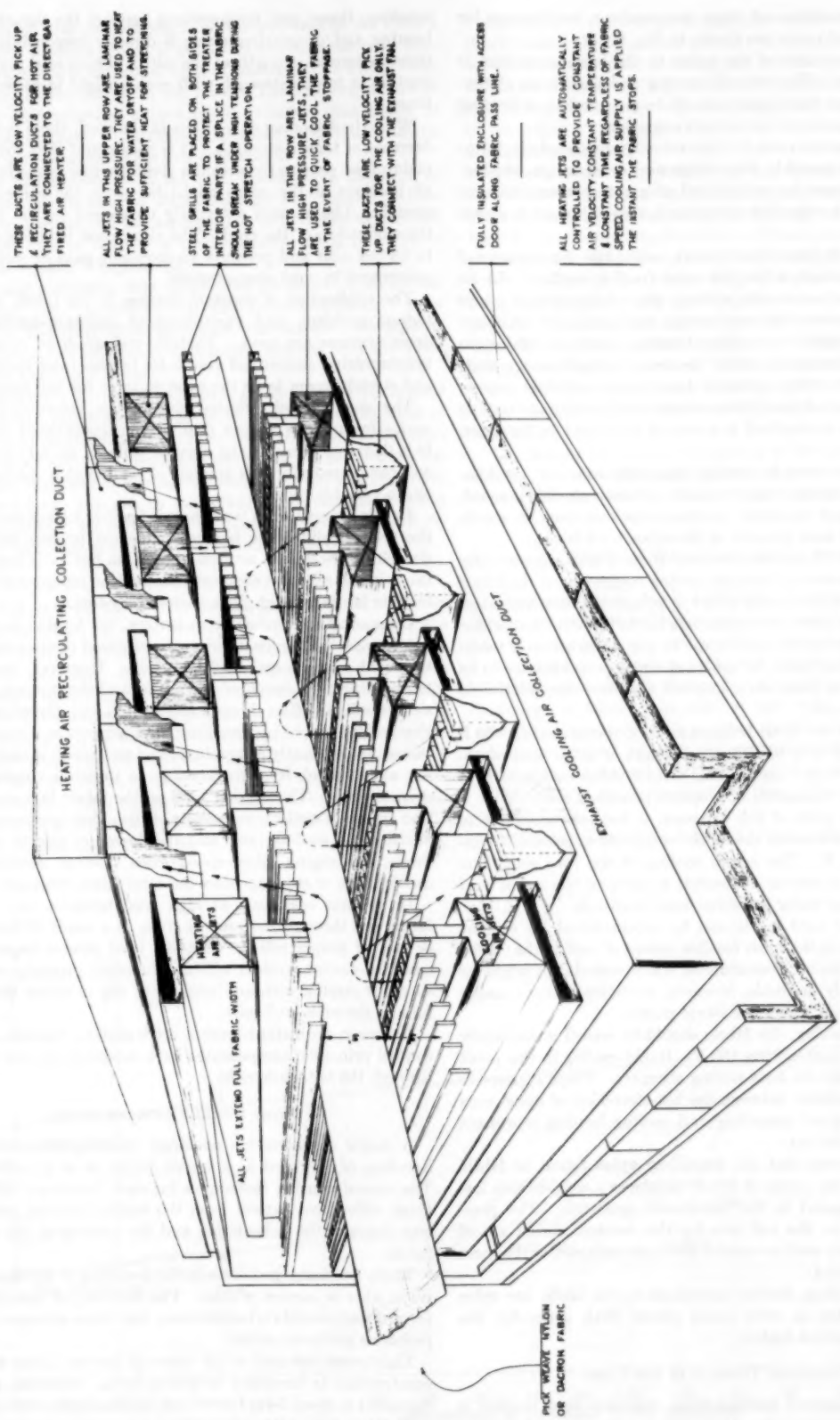
THESE DUCTS ARE LOW VELOCITY PICK UP & RECIRCULATION DUCTS FOR HOT AIR. THEY ARE CONNECTED TO THE DIRECT GAS FIRED AIR HEATER.
ALL JETS IN THIS UPPER ROW ARE LAMINAR FLOW HIGH PRESSURE. THEY ARE USED TO HEAT THE FABRIC FOR WATER DRYOFF AND TO PROVIDE SUFFICIENT HEAT FOR STRETCHING.
STEEL GRILLS ARE PLACED ON BOTH SIDES OF THE FABRIC TO PROTECT THE TREATER INTERIOR PARTS IF A SPLICE IN THE FABRIC SHOULD BREAK UNDER HIGH TENSIONS DURING THE HOT STRETCH OPERATION.
ALL JETS IN THIS ROW ARE LAMINAR FLOW HIGH PRESSURE JETS. THEY ARE USED TO QUICK COOL THE FABRIC IN THE EVENT OF FABRIC STOPPAGE.
THESE DUCTS ARE LOW VELOCITY PICK UP DUCTS FOR THE COOLING AIR ONLY. THEY CONNECT WITH THE EXHAUST FAN.
FULLY INSULATED ENCLOSURE WITH ACCESS DOOR ALONG FABRIC PASS LINE.
ALL HEATING JETS ARE AUTOMATICALLY CONTROLLED TO PROVIDE CONSTANT AIR VELOCITY CONSTANT TEMPERATURE & CONSTANT TIME REGARDLESS OF FABRIC SPEED. COOLING AIR SUPPLY IS APPLIED THE INSTANT THE FABRIC STOPS.
HEATING AIR RECIRCULATING COLLECTION DUCT
HEATING AIR JETS
ALL JETS EXTEND FULL FABRIC WIDTH
COOLING AIR JETS
EXHAUST COOLING AIR COLLECTION DUCT
PICK WEAVE NYLON OR DACRON FABRIC

FIG. 13 CUTAWAY PERSPECTIVE OF HOT-STRETCH UNIT

(Patents applied for.)

The optimum condition of time, temperature, and tension for nylon, cotton, and rayon are shown in Fig. 11.

The time of exposure of the nylon to the heating medium is also very critical. The rate of heating and conditions of tensions and exposure time apply equally to any method of heating that could be used in the hot-stretch operation.

The heating medium can be high-velocity convected air, infrared radiation, or possibly high-frequency or dielectric heating. The total heat must be maintained at a rate proportionate to fabric speed. A perspective view of a hot-stretch plant is shown in Fig. 13.

If the system is convection-heated, automatic dampering and volume control seems to be the most feasible method. In an infrared or a radiant-energy system, the relationship of fabric speed to the total electric heat energy available must be determined and controlled. A radiant-heating system is subject to all the physical laws which affect the use of radiant-energy heating. Color, reflectivity, distance from fabric-radiating source, skin coefficient and element temperature, and heating and cooling rate, all must be determined in terms of total heat in the nylon fabric itself.

Convection hot-stretch systems maintain constant air temperatures and constant impact velocity. Relative to fabric speed, convection systems maintain constant exposure time at conditions of constant heat transfer to the nylon-cord fabric.

In any hot-stretch system, constant time of heat exposure also requires the operation of suitable cooling equipment so the fabric can be cooled quickly in the event of complete fabric stoppage. If this is not done, the fabric remaining in the hot-stretch machine deteriorates very rapidly and its use in tire construction is undesirable. The fabric must be cooled at a rate proportionate to its change in speed to maintain a constant exposure time relative to its decreasing speed.

Exposure of 60 sec at air temperatures of approximately 400 F will cause a rapid drop in ultimate strength of nylon cord to approximately 20 lb per cord. This tensile value compares to a tensile of 27.5 lb obtainable in properly processed cord.

At the critical point of full exposure, it becomes necessary to reduce the temperature in the fabric as quickly as possible to approximately 150 F. The actual cooling of the cord should be accomplished in a matter of seconds to prevent the nylon cord from deteriorating under excessive heat conditions.

Cooling of the cord fabric can be accomplished by various means. Cold air is the most feasible means of cooling the fabric, although, under some circumstances, water-cooled rolls might be used. It is highly desirable, however, to minimize the number of roll surfaces over which the fabric passes.

After hot-stretching, the fabric should be cooled immediately to temperatures approaching 150 F. Rapid cooling to this point seems to terminate the heat-setting abruptly. There appears to be a distinct similarity between the hot-stretching of nylon cord and usually accepted annealing and cooling heating treatment of ferrous wire and rod.

It has been found that the wound-up nylon fabric, at fabric temperatures in the order of 200 F, produces a considerable loss of the stretch applied in the hot-stretch operation. The pressures developed on the roll core by this residual shrink are of such an extent that steel or wooden shells are collapsed or the cores are actually crushed.

After quick cooling, further operations on the fabric are quite generally the same as with rayon except that, generally, the tensions are somewhat higher.

Constant Tension in the Cord

Along with constant heating time, constant cord tension is of equal critical importance. Data showing the effects of varying tensions, times, and temperatures support the theory that overheating and/or overtensioning is directly responsible for reductions in tensile strength of the cord. There is a very rapid reduction in tensile strength with even a slight increase in heating time.

With the lessening of tensions in the cord, there is a less rapid decrease of tensile strength. It is paramount that the tension be maintained constantly within a given narrow range in order that all portions of the cords might be given the same amount of stretch. The tension externally developed in the cord causes the elongation of the cord. The elongation is in the order of 14 to 18 per cent and presents an economic gain as well as an improvement in cord characteristic.

The application of constant tension to the fabric presents no serious problem, and variable-speed, constant-tension electric-drive systems are used. Variable-voltage-drive systems, using synchronizing dancer roll stands for tension (and therefore speed and stretch) seem to be the most suitable for this operation.

The cord stretch effected during the hot-stretch operation varies from 15 to 25 per cent of the unstretched cord length. It is therefore necessary to provide suitable electric drives which will compensate for this amount of stretch at a constant tension and a variable speed.

In this discussion on tension, it is feasible to mention in passing the relative difference between constant tension and constant stretch. It is not an academic question but it is one of importance in the selection of suitable drives and equipment for propelling the fabric through the hot-stretch system.

On a constant-stretch drive system, the tension in the cord is determined by the fixed difference of speed relationship between the front squeeze and back-pull rolls. However, the operating temperature, the speed of the fabric, and the slippage at the roll stands, all will affect directly the stretch and, therefore, the cord characteristic. As tension, time, and temperature must be maintained as constantly as practical, and as tension is easier to measure and control, it seems feasible to utilize a constant-tension drive system. The tension load on the fabric in a constant-tension system is uniform regardless of the nylon temperature during heating and cooling cycle and the degree of stretch in the cord fabric. Changing fabric-speed rates through acceleration and deceleration or stoppage does not alter fabric tensions.

In systems operating at high cord tensions, the amount of fabric slip through the squeeze rolls, as a result of fabric tension or lack of proper roll-nip capacity, is of utmost importance. A constant-stretch system will not function properly at constant speed or stretch with any amount of slip in either the holdback rolls or the main-pull rolls.

However, a constant-tension drive system, because of its basic control principle, compensates for a moderate degree of slippage through the holdback rolls.

Some Other Considerations

A major consideration on larger treating installations is the handling of the cord as a woven fabric or as a weftless fabric. The manufacturing techniques for each fabric are different and those differences extend from the textile-finishing plant all the way through the calendering and the windup of the rubberized fabric.

There are many problems in the handling of weftless-cord fabric in wide or narrow widths. The economy of operation points toward weftless-fabric installations, but there are many operating problems yet to be solved.

Eighty-one per cent of all types of carcass fabric used for tire construction is furnished in woven form. Nineteen per cent is furnished in spool form for creel or weftless operation.

Weftless full-width nylon fabric for creel operation, for latex-

ing hot stretch and calendering is not as yet used in the United States. One problem which is yet to be solved in the creel handling of nylon cord is the proper type of package that will provide for easy continuous payoff in the creel. The slippery nature of the nylon cord provides a problem in winding that nylon in stable cone packages. Continuous tied end creel letoff can be done only with cone packages properly wound.

Another major problem in the creel operation is the type of knot which must be used to withstand high cord tensions. Barrel knots will withstand the tension, but it is doubtful whether barrel knots can be tied by a creel operator while the cord is traveling at relatively high speeds.

It has been suggested that nylon cord be provided on beams and paid off into the hot-stretch unit from this beam. It is doubtful, at this stage of development, whether the technique of beaming nylon has progressed sufficiently to allow for high-speed payoff from individual beams.

The peculiarity of nylon to shrink under certain conditions presents some problems in high-speed beaming. Shrinking of some cords on the beam tends to force these cords to slip between adjacent cords and into previously wound layers of cord on the beam. The operation of paying out smoothly from a beam, under these conditions, has yet to be solved. Continuous treating operations cannot be performed continuously from a beam regardless of the lengths of the cords wound on the beam.

Proper Splicing

As 80 per cent of the tire carcass cord fabric is furnished in woven construction on rolls, continuous operation requires the splicing of roll ends together to form a continuous web through the hot-stretch operation. In the operation of a calender-train unit or a separate machine, the need for vulcanized splices to withstand hot-stretch tensions is of paramount importance.

On a separate latex-and-hot-stretch operation a simple lap splice made with suitable adhesives and at pressures and times that will cure the splice properly is already worked out. Sewn splices have been used at tensions to 4000 lb, but the time consumed in sewing presents a problem. Vulcanized splices are in use today at tensions of 12,000 lb.

Splices for a calender-train operation are somewhat more difficult because the total thickness of the splice must not exceed the total thickness of the rubberized and calendered fabric. If a total thickness of the splice is greater than the thickness of the rubber and fabric, the calender rolls must be parted to allow for passage of the splice. This results in a production loss and scrap fabric and must be avoided.

Splices are now being worked on that should prove suitable for operations at 20,000 lb and fabric elongations of 20 per cent. Pressures of the platen press are in the order of 1000 psi, the curing times are in the neighborhood of 75 sec, and the temperatures of the platens are in the neighborhood of 500 F.

After calendering, the handling of nylon fabric presents no particular problem other than rayon, with one exception. This principal exception is the tension value of the cord prior to and immediately after calendering due to the shrinking tendency of nylon as it is being calendered with hot rubber stocks. The latest trains that are being designed and built for nylon cord are built to develop pre and postcalendering tensions of up to 2000 lb maximum.

The gage of the fabric, the spacing of the cords, and the length stability of the rubberized fabric are most uniform when the cord is calendered at the correct high tensions.

2000-End Versus Smaller Number of Ends

The economics of the hot-stretching operation point directly to the handling of full-width fabric in woven form rather than the handling of small numbers of individual ends.

The cord processed in a woven fabric, 2000-end machine, although operating at slower linear speeds, will produce more poundage of fabric than smaller end units operating at high speeds. A machine with a rated continuous capacity of 7½ yd per min (ypm) will turn out approximately 125 tons of fabric per month. A 50-end unit, operating at approximately 85 ypm, will produce in the same length of time approximately 25 tons of latexed cord.

In the handling of the full-width fabric, the problems of draw-

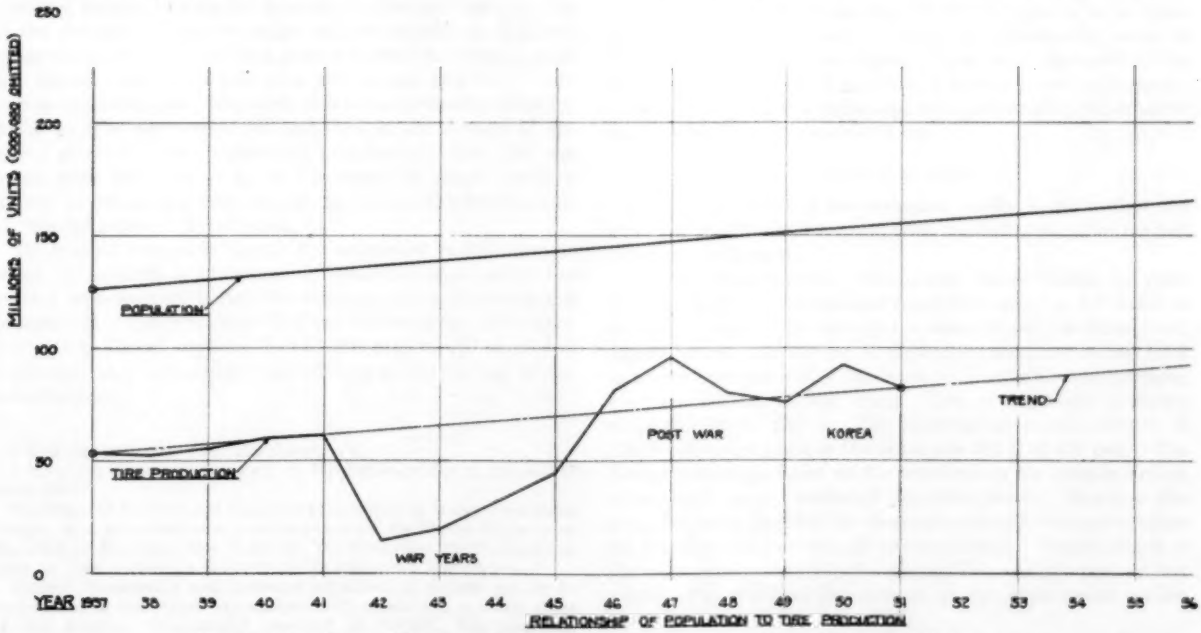

Fig. 14 Long-Range Trend of Population of United States From 1937 to the Present

down, cord misalignment, and uneven cord count have been solved suitably. The loss in fabric width on a machine operating at 10,000 lb tension on a 28-cord-per-in. fabric amounts to approximately 5 per cent of the width of the full fabric. In other words, a 60-in. fabric will latex and hot-stretch and can be wound up at an over-all width of approximately 57 in.

A $7^1/_2$-ypm machine can be operated easily by two men and, likewise, a machine with a capacity of 30 ypm can also be operated by two men. Machines with higher capacities require proportionately more attendant labor.

Cost of Latexing Fabric

Cost studies indicate that the cost of latexing full-width fabric runs approximately 5 cents per yd of nylon latexed and hot-stretched. The cost of operation of latexing and stretching nylon is not appreciably greater than the cost of latexing and tension-drying rayon.

In handling cord in smaller numbers of ends, the cost of respooling, beaming, and the extra attendant labor at the creels indicate that the cost of latexing per yard of 2000-end weftless fabric is approximately 15 cents per yd.

The problem of hot-stretching nylon is now more thoroughly understood, and as newer and large installations are completed, further advanced techniques are being applied to the design of hot-stretching equipment.

There is no doubt in the author's mind that through co-operation among the manufacturer of cord-fabric materials, the manufacturer of tires, and the designers of equipment that the problems of hot-stretching for all cord-fabric materials will be more thoroughly understood. When that day arrives, the rubber industry can take full benefit of all these synthetic materials.

In order to show some of the long-range possibilities for rayon and nylon as tire-cord material, we refer you to Fig. 14. This illustration shows the long-range trend of the population of the United States as it actually existed from 1937 to the present day. Also shown is an extension of the curve to the year 1956.

Also plotted to the same yearly ordinance is the actual tire production in terms of millions of casings. The trend line, drawn through the tire-production curve, indicates the long range and close parallel between the rates of tire-casing production and long-term population increase.

Although these figures are available until 1970, the illustration only goes to 1956.

Barring sudden or major changes in conditions, it seems feasible that tire production would increase in the same proportion as population. Even with greater mileage, the general increase in the use of rubber-tired vehicles for all services should compensate for longer mileage in original equipment tires.

These curves, we believe, most graphically depict the challenge of the future that lies ahead for the rubber industry.

Action of Boiler Water on Steel—Attack by Bonded Oxygen

By C. E. KAUFMAN,[1] W. H. TRAUTMAN,[1] AND W. R. SCHNARRENBERGER[1]

This paper includes information relating to attack by boiler water on steel of boilers and superheaters. Emphasis is placed on the power of water itself, in the absence of dissolved oxygen, to oxidize metal, causing substantial damage under some conditions. Examples of aggressive action drawn from field experience are explored by metallographic study of damaged specimens. Laboratory results show that hydrogen embrittlement can be produced and controlled in test bombs. It is concluded that the power of bonded oxygen of water to react with steel can be curbed by making both chemical and physical changes in boiler systems.

Introduction

CORROSION of boilers dates back to the days of James Watt. In all the time since then, we have been faced with problems which really have not essentially changed in nature but which have grown increasingly serious, or so it probably seems to us from our particular vantage point in time. With higher pressures, greater capacities, and rigorous standards of availability, we rightfully feel that we always must be improving our knowledge of the diagnosis and cure of corrosion in boilers.

Attack on metal shows up in a variety of ways. Can we, out of a wealth of somewhat dismaying data, establish any unifying principles to chart our course?

Fatigue and stress-corrosion exact their toll of cracking. Caustic embrittlement produces intergranular failure at riveted seams and rolled-in tube ends. Much damage has been done by dissolved oxygen, but the development of efficient deaeration and of the chemical oxygen scavenger sodium sulphite in comparatively recent times has led to a great reduction in corrosion from this source. In only a few cases will oxygen still be at work during operation; more frequently it is to blame during stand-by. Although it is also recognized that even in the absence of dissolved gaseous oxygen, substantial oxidation of boiler steel can occur, there still appears to be a tendency to blame dissolved oxygen for difficulties which are directly attributable to the action of "bonded oxygen" (1)[2] of water.

By bonded oxygen is meant the oxygen of constitution in water. Compared to the traces of dissolved oxygen which can enter a well-operated boiler, the reservoir of bonded oxygen is tremendous. Water contains 88.9 per cent oxygen. Of course, it is just as absurd to claim that all this oxygen will react with boiler steel as it is to assign responsibility for all damage to dissolved oxygen.

The water-metal reaction can be summarized as

$$4H_2O + 3Fe \rightarrow Fe_3O_4 + 4H_2$$

The exact mechanism of the reaction and the role of possible intermediate compounds involving ferrous, ferric, and ferrate-type ions are not completely clear. Most boilers operate without any trouble because this reaction of bonded oxygen, proceeding very slowly, maintains a thin protective film of iron oxide.

One factor which may lead to serious attack by bonded oxygen is the effect of boiler water in which dissolved material has been concentrated under the impetus of high heat input, excessive temperature, inadequate circulation, or action in capillary crevices of deposits adjacent to metal. When boiler water is thus concentrated and rinsing action by the more dilute average water is not consistent or immediate, the ordinarily protective layer of iron oxide, very thin to begin with, may become unstable and dissolve or slough off. Any free caustic or acid in the dilute water will concentrate and contribute greatly to destruction of the oxide film. Once the layer is gone, boiler water can react with the fresh steel surface to produce more iron oxide and hydrogen.

Sometimes when a mildly alkaline boiler water contains no free caustic, appreciable attack by bonded oxygen takes place. Under these circumstances it appears that there is not enough alkalinity to maintain a protective oxide barrier.

It seems unnecessary to assume that boiler water breaks down to form gaseous hydrogen and oxygen before reaction takes place since temperatures must be quite high to produce significant decomposition. Even at 1200 F and 1400 psia, oxygen derived from breakdown of steam will be less than 0.01 ppm (2). Direct action of water or steam on metal therefore appears to be likely in most situations. Steam reaction on overheated metal is illustrated by the adherent layers of iron oxide produced on the inside of low-carbon-steel superheater tubes exposed to temperatures of 900 to 1000 F or higher and by attack on alloy superheater tubes at higher temperatures (3, 4).

Case Histories

Several case histories are presented briefly to show what has happened at different locations when bonded oxygen has reacted vigorously with steel.

Marine Desuperheaters. Steam was desuperheated by passing it through low-carbon-steel loops submerged in the water of the steam drum. The inlet line entered above the water level, dipped beneath the surface of the boiler water, and looped back and forth from one end of the drum to the other before emerging near the bottom of the drum. This arrangement is shown schematically in Fig. 1. The superheated steam entered at 750 F and temperature in the drum was 460 F at 450 psig. The desuperheater line failed at the interface on the outside surface where boiler water contacted saturated steam. Nowhere else along the entire length of the desuperheater either above or below the interface was any sign of corrosion found. Severe attack at the water level occurred in the original line and in a replacement nipple. Fig. 2 shows the damage to the replacement section which had been welded into place.

Tests for dissolved oxygen in the feedwater by the Winkler

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[1] Hall Laboratories, Inc., Pittsburgh, Pa.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Joint Research Committee on Boiler Feedwater Studies and presented at a joint session with the Power Division at the Annual Meeting, New York, N. Y., November 29–December 4, 1953, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, November 27, 1953. Paper No. 53—A-241.

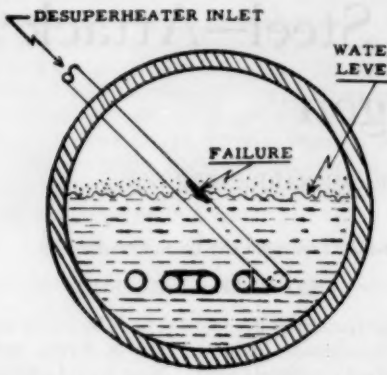

FIG. 1 DESUPERHEATER IN STEAM DRUM

procedure consistently gave zero results. This meant that dissolved oxygen in the feedwater was less than a maximum of about 0.05 ppm. Because of the strong tendency of gases to escape with the steam, the amount of oxygen remaining dissolved in the boiler water splashing up on the tube is believed to be so low as to be zero for all practical purposes. The absence of any sign of corrosion elsewhere on the desuperheater line or on the drum surface further indicates lack of dissolved oxygen.

A run was also made with enough sulphite added to provide a reserve after reaction with oxygen. Damage developed at the same rate as when sulphite was absent.

The mechanism of attack would appear to be as follows. The temperature difference or Δt between the 750 F steam inside the desuperheater and the 460 F boiler water outside is 290 deg F. Allowing for some temperature drop in the tube wall, a substantial temperature difference still remains. As boiler water splashes and swirls against the desuperheater tube at the water line, a film of boiler water will concentrate intermittently against the steel. With free caustic alkalinity maintained in the boiler water, as concentration took place, the strong NaOH solution produced helped remove protective iron oxide and attack by water proceeded. Although fresh and unconcentrated boiler water presumably would intermittently replace the concentrated film, the constant "chemical slugging" at this critical spot finally produced failure.

In another marine installation with a similar desuperheater, it was learned that following failure of a carbon-steel tube, a nipple made of 347 stainless steel had been used as replacement and had failed at the water level. Fig. 3 shows the section of stainless-steel tube removed after failure. Wastage here was not so prominent but many transgranular cracks developed. In the case of this alloy, the repeated variation in temperature as water washed up on the hot metal surface may have been more important than in the case of the low-carbon steel.

In marine desuperheaters of this kind, relief from damage has been obtained by welding on a sleeve at the point of attack. In some units the flow of superheated steam is in the reverse direction. The steam line enters under the surface of the boiler water and by the time it emerges at the water line, temperatures of steam and boiler water are close enough together so that no damage is done.

Trouble at Welds. A high-capacity boiler in a central station operating at about 1600 psi had suffered no particular difficulty during the first 16 months of operation. Then two studded tubes were found to be leaking in the slag screen separating the primary furnace from the secondary furnace. Both leaks occurred about 7 ft above the floor on the side facing the primary furnace at a point where welds had been made with internal chill

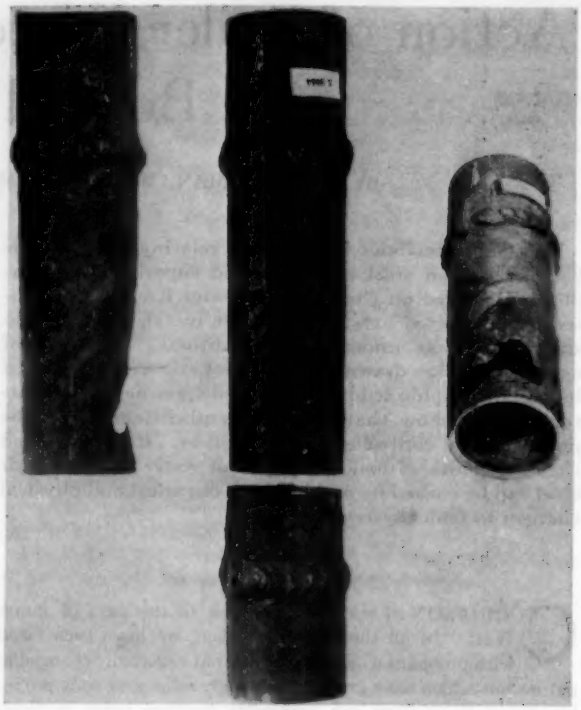

FIG. 2 DAMAGE TO DESUPERHEATER REPLACEMENT SECTION

FIG. 3 ATTACK ON STAINLESS-STEEL DESUPERHEATER SECTION

rings. The studs had been worn down in the area of attack. Fig. 4 shows one of the tubes split open and a small pile of material from the inner surface. A piece of chill ring projecting from the center of the right-hand section may have been bent out to this position during sawing of the tube. However, it is obvious that the chill ring had not been in good contact with the tube wall and that corrosion had occurred in and adjacent to a pocket formed between the ring and the wall. Black material, largely magnetic iron oxide and copper with only small amounts of other substances, was concentrated on the internal surface of the section in the area of the chill ring. Adjacent sections were essentially free of deposit.

Metallographic examination showed that the steel had been markedly overheated independent of any change produced when the studs were placed on the tube. At the left in Fig. 5 is an area about one third of the way through the wall from the outside surface of the tube. Here the metal has been quenched from a temperature of about 1400 F. Dark-colored carbides are concentrated at the grain boundaries. The outside surface of the tube at the point of principal attack had been essentially decar-

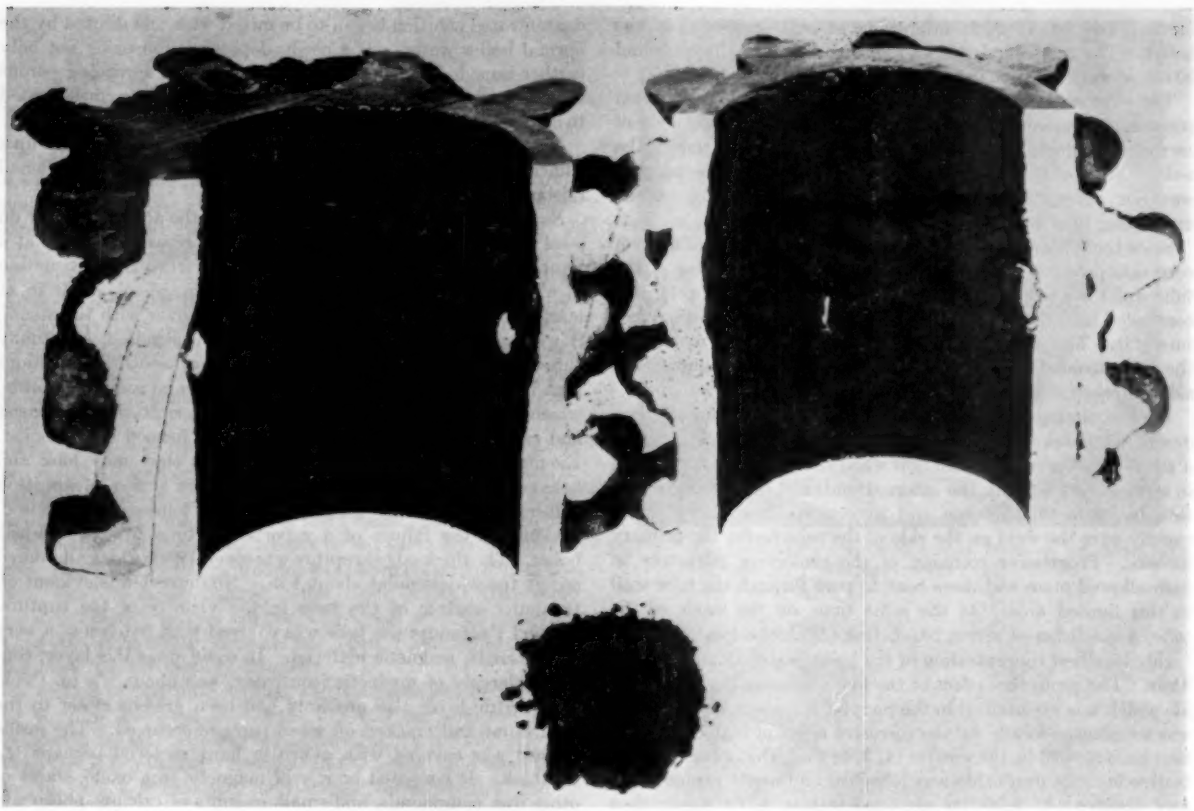

FIG. 4 CORROSION AT WELD AND PILE OF MATERIAL FROM INNER SURFACE

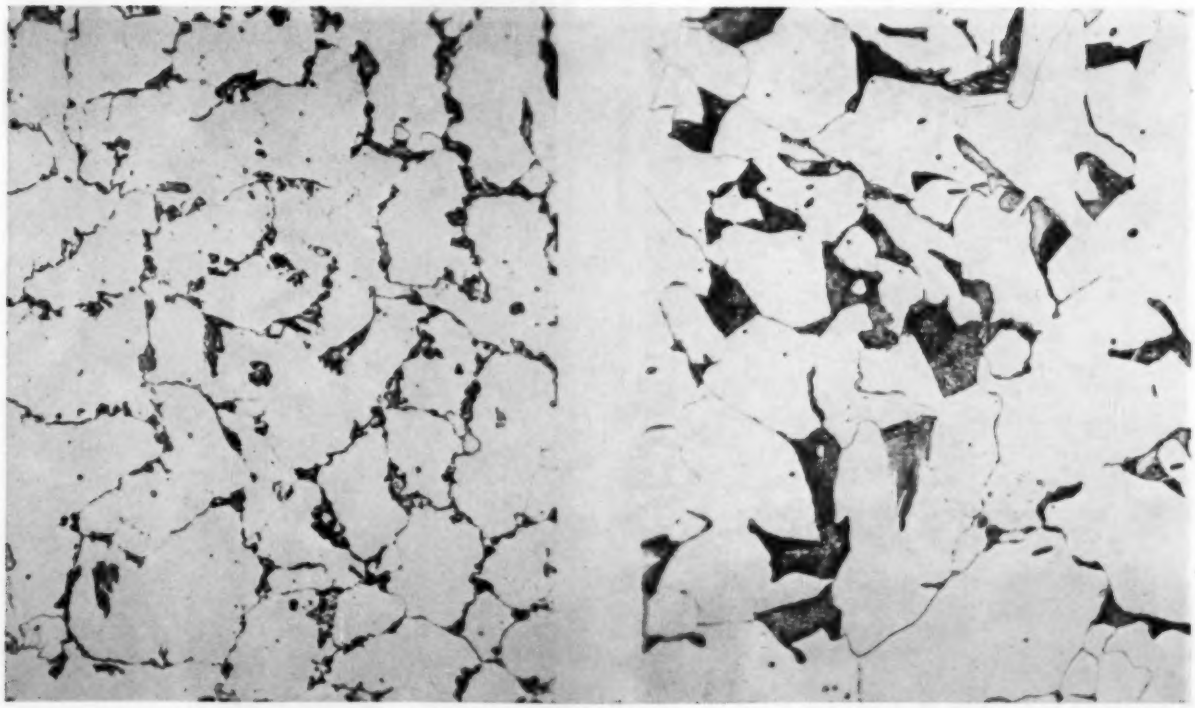

FIG. 5 MICROSTRUCTURE AT WELD; ×500

(*Left:* From failed area, showing overheating. *Right:* From side of tube, showing normal steel.)

burized and considerable intergranular oxide penetration was noted. The right-hand side of Fig. 5 shows normal steel found 90 deg around the tube from the failure.

The overheating in the localized area where failure occurred presumably was caused by accumulation of an insulating layer of corrosion products beneath and adjacent to the chill ring of the weld. Most of the magnetic iron oxide comprising the mass of corrosion products obviously was derived from attack on the underlying tube surface.

Since there is no evidence of internal attack, no accumulation of corrosion products, and no evidence of gross overheating of the tube wall for a foot above and below the failure at the weld, it is concluded that the critical factor must be the weld or some feature of it. The presence of the chill ring, incompletely bonded to the tube surface, appears to be the basic cause of failure. Residual stresses at the weld also would tend to localize the damage.

Reconstructing the probable course of events from the evidence, it is believed that the crevice under the chill ring provided a steam-filled barrier to heat flow when the boiler was first placed in service. As a result, the external studs and chrome-ore insulation ran somewhat hotter and were worn down a bit more rapidly over the weld on the side of the tube facing the primary furnace. Progressive thinning of the protective refractory in turn allowed more and more heat to pass through the tube wall in this limited area. At the same time, on the inside of the tube, a condition of steam blanketing (5) in the crevice caused highly localized concentration of the boiler water almost without limit. The protective effect of the low concentration of hydroxide which was maintained in the body of the normal boiler water was supplanted locally by the corrosive effect of high concentration on hot steel in the crevice (1, 2, 6, 7). The ordinarily protective layer of iron oxide was breached and newly exposed iron was dissolved in this film of concentrating boiler water, then carried out and redeposited as black magnetic iron oxide where the concentrated solution began to be mixed with and diluted by the normal boiler water. The newly deposited iron oxide not only further impeded heat flow but also provided a spreading porous blanket within which normal boiler water would be concentrated to attack more of the tube surface.

Finally, when the internal attack of the tube wall had built up a sufficient insulating layer of iron oxide, the tube formed a small blister and failure ensued.

Superheater Tube. In the boiler proper the action of water on steel seems most easily explainable on the basis of reaction at a liquid-solid interface. When attention is turned to the situation in a superheater, it seems likely that damage can occur in at least two ways. If boiler water is carried over with the steam, it will tend to concentrate as temperature is raised, eventually throwing material out of solution. Very concentrated solutions high in such materials as sodium hydroxide and sodium chloride, which have increasing solubility with temperature, will persist and can act in a manner similar to those formed in the boiler. Eventually, as these "solutions" dry out they may look and behave more like solids. Besides water-film action, steam itself will react at an increasing rate with steel as temperature is raised.

Consider the failure of a superheater tube from a 425-psig boiler. The thick-edged rupture was several inches long and flared out at the widest point about 1 in. No deposit was evident on the outer surface of the tube in the vicinity of the rupture. Beyond the failure the tube was covered with patches of a very hard, brittle, magnetic material. In some areas this layer, consisting largely of magnetic iron oxide, was about $^1/_{32}$ in. thick. This brittle layer also probably had been present closer to the failure but had cracked off when rupture occurred. The inside surface was covered with a brittle hard material perhaps $^1/_{32}$ in. thick. It consisted largely of magnetic iron oxide, traces of other iron compounds, and small amounts of calcium phosphate and burkeite ($Na_2CO_3 \cdot 2Na_2SO_4$).

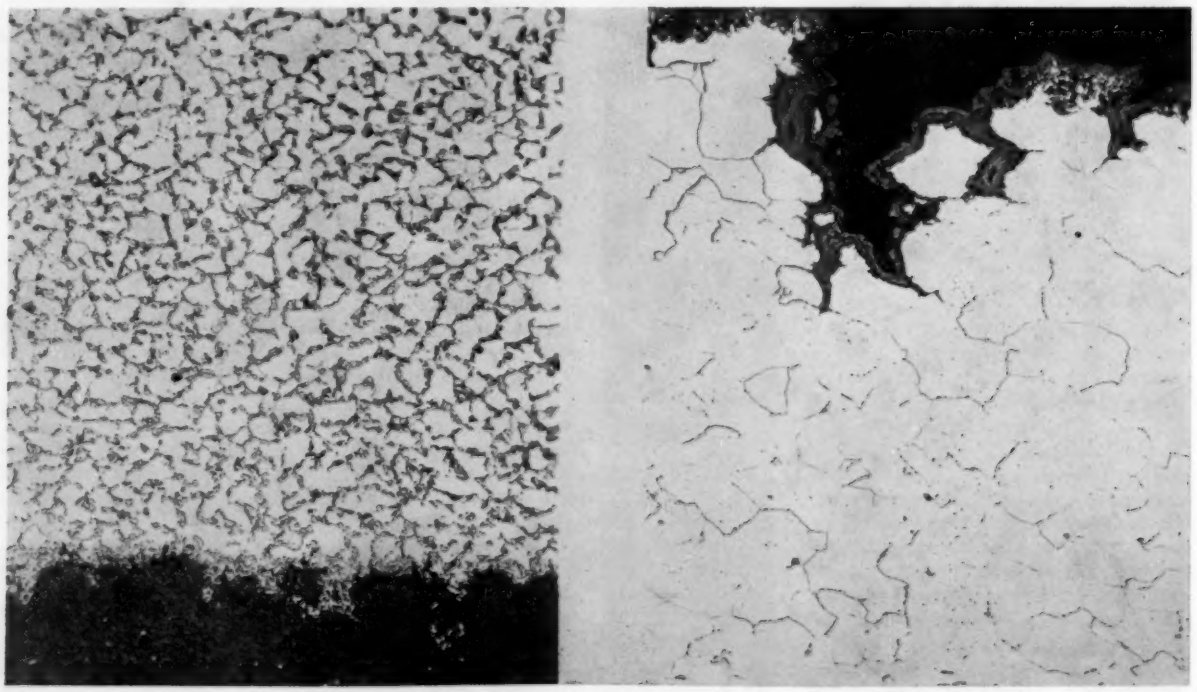

FIG. 6 MICROSTRUCTURE OF FAILED SUPERHEATER; ×250
(*Left:* Inside surface. *Right:* Outside surface.)

The original metal structure presumably had been characteristic of low-carbon steel. However, Fig. 6 shows that overheating had been drastic at the inside and outside surfaces of the tube at the failure.

"Burning" was confined to this one tube in the superheater, so it would seem that some peculiarity of heat input or circulation was primarily accountable for the failure. Attack was by excessively heated steam, with a small amount of entrained boiler water perhaps contributing to damage.

Hydrogen Embrittlement. After several years of operation, screen tubes failed in a 1400-psig high-capacity boiler. This unit had been operated with little or no free-caustic alkalinity and with oxygen scavenger added. The break in one tube was located a few feet beyond a lower or mud drum where the tube was inclined only slightly to the horizontal. The force of the rupture whipped the tube so that another break was produced even closer to the drum.

The failure was thick-edged, although near the edges of the torn metal the wall had been thinned about $^1/_{16}$ in. as a result of attack on the inside surface at the top of the tube. Fig. 7 shows part of the failed tube. Gouging was noted at the inside surface of an unfailed portion of the section on the same side as the failure. Very little corrosion product was found on the inside of the tube; this was largely magnetic iron oxide (Fe_3O_4), with some hematite (Fe_2O_3), a little copper, and traces of other material. When the doubled-up portion was opened subsequently and bent back, the tube wall broke apart.

A section of the tube, cut off and macroetched, is shown in Fig. 8. The darkened area at the top, starting from the inside surface, indicates hydrogen embrittlement (8, 9, 10, 11, 12). When metal from several locations in the failed tube was examined under the microscope, the patterns varied widely. Fig. 9

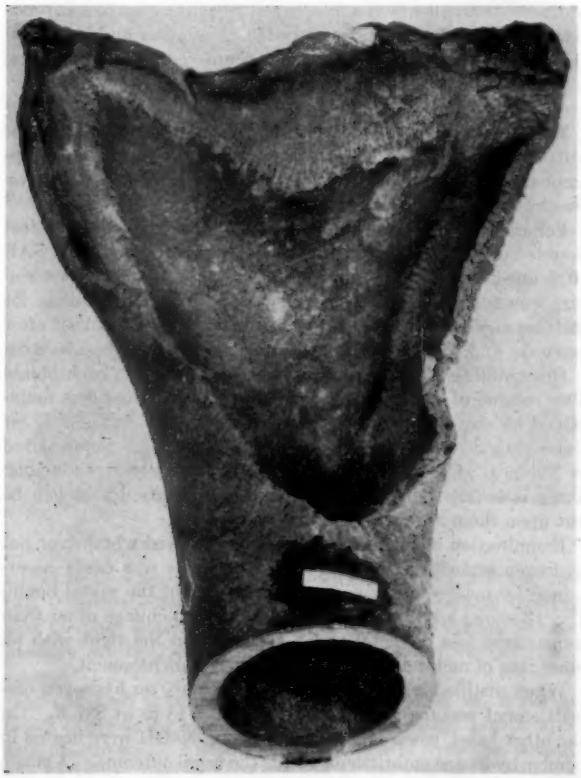

FIG. 7 FAILED SCREEN TUBE

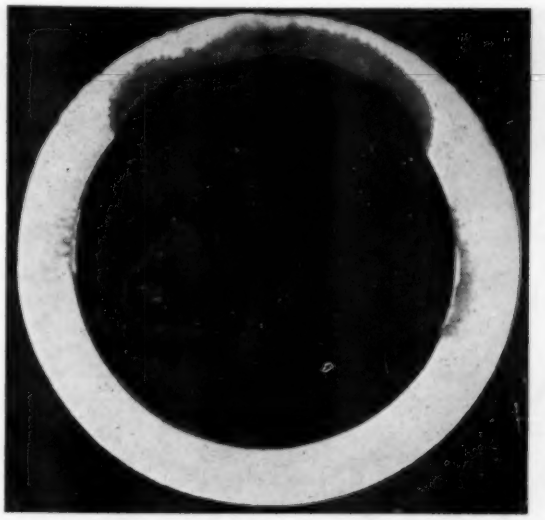

FIG. 8 MACROETCHED SECTION SHOWING DAMAGE BY HYDROGEN EMBRITTLEMENT

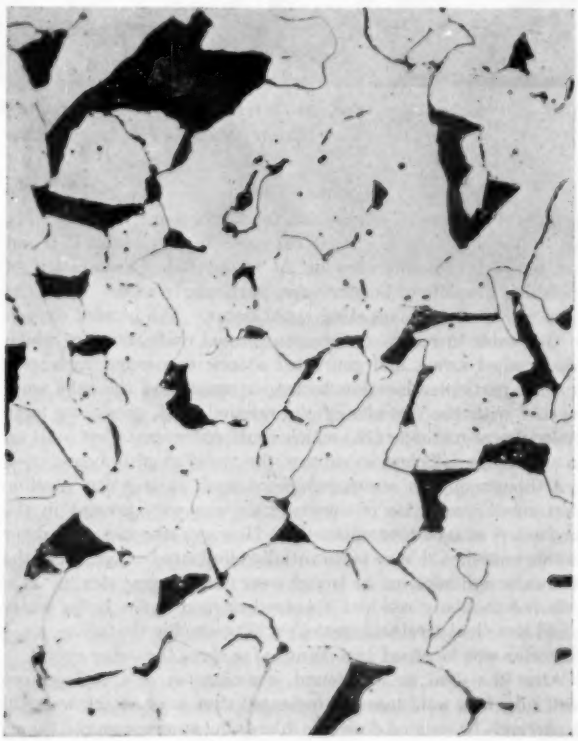

FIG. 9 NORMAL MICROSTRUCTURE AT OUTSIDE SURFACE OF TUBE 180 DEG AROUND FROM FAILURE; ×500

shows essentially undamaged metal at the outside surface of the tube 180 deg around from the failure. Fig. 10 shows an area near the inside surface of the specimen taken at the failure before and after etching. The original structure at this location has been destroyed, with little if any carbide left, and with many intergranular cracks penetrating the metal. This appearance is characteristic of other cases where hydrogen embrittlement has been found.

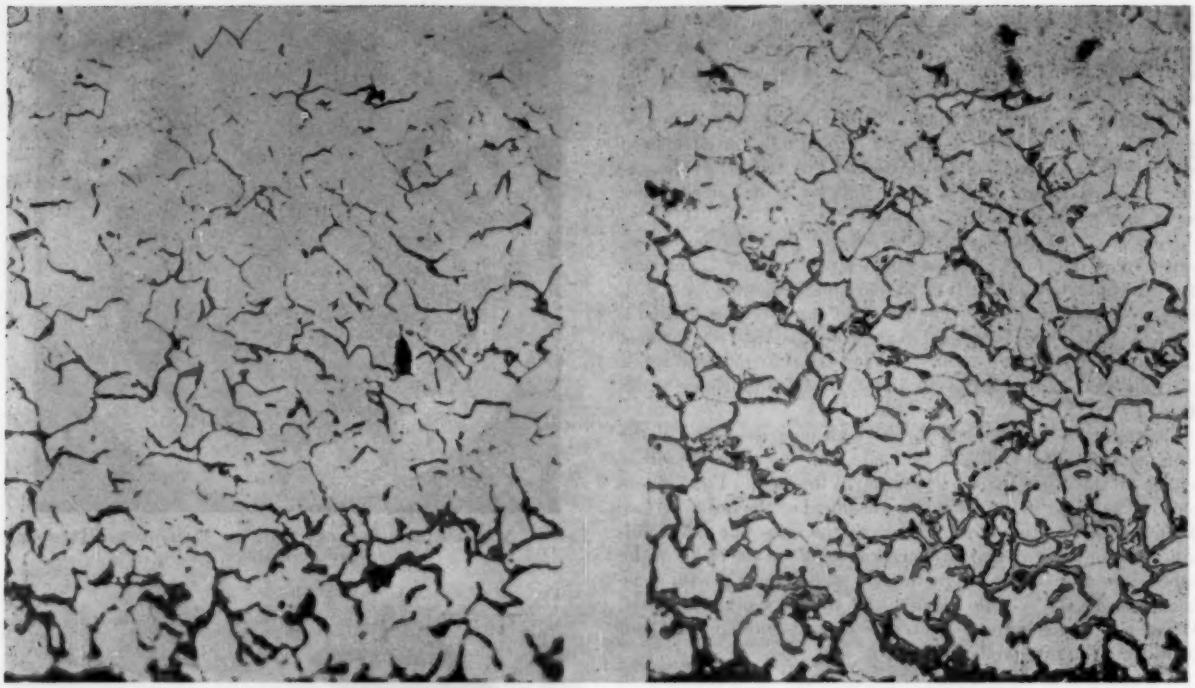

FIG. 10 MICROSTRUCTURE AT FAILURE SHOWING HYDROGEN EMBRITTLEMENT; ×500
(*Left:* At inside surface of specimen before etching. *Right:* After etching.)

Action to produce this deterioration of the metal can be visualized as a sequence in which the temperature intermittently or continuously exceeded that corresponding to saturation, and the temperature difference or Δt forced the concentration of splashing droplets of boiler water, particularly at the top of the tube where steam-blanketing could occur. The bonded oxygen of the water in reacting with iron formed oxide, most of which was washed away, and generated atomic or nascent hydrogen. At this particular location hydrogen penetrated the tube wall, reacted with the carbides of the original steel, producing large molecules of methane (13) which could not migrate but built up to a pressure sufficient to rupture the metal at grain boundaries.

Although oxygen scavengers were used, it is quite possible that small quantities of dissolved oxygen were present in the feedwater, at least intermittently. However, the steam scrubber in this unit should have substantially eliminated oxygen from the feedwater spilling from its trough over the emerging steam. It is believed that any residual dissolved oxygen in the boiler water could have had no significant effect in producing the failure, since corrosion was localized at a point so far along the boiler circuit.

After the damage was found, examination of a replacement wall tube at a weld location indicated that some attack was still occurring. It seemed desirable in arriving at optimum conditions to increase the NaOH content of the water from essentially zero to the range of 30 to 50 ppm. Caustic content is a particularly critical factor, since neither a deficiency nor an excess will permit a protective layer of oxide to be formed (14). The complexity of this adjustment is certainly not reduced by the fact that on the one hand we have the average relatively dilute boiler water and on the other the constantly changing concentrated boiler water produced where temperature exceeds saturation value.

At about the same time as alkalinity was adjusted, spirals were installed in the affected tubes, permitting a more thorough and even wetting of the internal surfaces. On general principles, a reserve of sodium sulphite was carefully maintained (15). In addition, the boiler was acid-cleaned to remove corrosion products which might have interfered with circulation and heat flow.

No further difficulties have been encountered for about 3 years.

LABORATORY EXPERIMENTS

Tests run in our laboratories have reproduced hydrogen embrittlement (16). Fig. 11 is a schematic drawing of the equipment used to produce the requisite temperatures and pressures. Bombs and parts are shown in Fig. 12.

For ease in manipulation, the bombs served not only as test vessels but also as specimens. They were fabricated from SAE 1010 low-carbon steel. A simple plug and following-screw closure was adequate and no special provision had to be made for their removal since the bomb was merely allowed to cool off after exposure and was then sawed through perpendicular to its axis.

One milliliter of water or solution was added to each bomb. The volume of the bombs was 2.3 ml. The furnace was maintained at any desired temperature between 392 and 932 F for times ranging from about 3 to 72 hr. Results are summarized in Table 1. On the whole, the tests were fairly reproducible; hence it is felt that results and the interpretation that can be put upon them are meaningful.

Examination of metal following a test showed whether or not hydrogen embrittlement had occurred. This was easily ascertained by macroetching a slice or doughnut of the sawed bomb. Fig. 13 shows a specimen on the left with no change other than some slight loss of metal and a specimen on the right with an inner ring of metal damaged by hydrogen embrittlement.

When distilled water was added to a bomb, no hydrogen embrittlement was found after an exposure of 18 hr at 887 F. On the other hand, when solutions containing NaOH were heated in bombs, hydrogen embrittlement was the usual outcome. A single steam-water phase existed in the bomb to which only water was

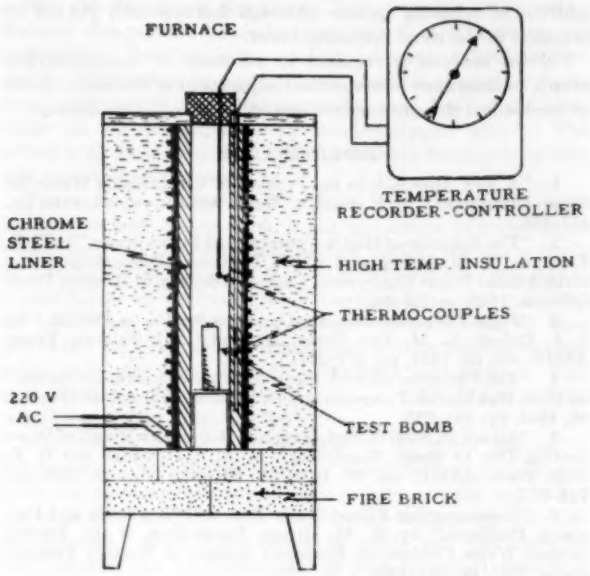

FIG. 11 TEST EQUIPMENT

TABLE 1 HYDROGEN EMBRITTLEMENT IN LABORATORY BOMBS

Test No.	Bomb charge	Temperature, deg F	Time, hr	Results[a]
1	Distilled water	887	18	–
2	1 ml 40% NaOH solution	887	72	+
3	1 ml 40% NaOH solution	887	18	+
4	1 ml 40% NaOH solution	887	3	+
5	1 ml 40% NaOH solution	932	18	–
6	1 ml 40% NaOH solution	932	18	+
7	1 ml 40% NaOH solution	932	66	+
8	1 ml 20% NaOH solution	932	18	+
9	1 ml 40% NaOH solution	392	18	–
10	1 ml 40% NaOH solution	392	18	–
11	1 ml 40% NaOH solution + 0.1 g Na_2SO_3	887	18	+
12	1 ml 40% NaOH solution + 0.23 g $NaNO_3$	932	18	–
13	1 ml 40% NaOH solution + 0.23 g $NaNO_3$	932	72	–
14	1 ml 40% NaOH solution + 0.115 g $NaNO_3$	932	18	–
15	1 ml 40% NaOH solution + 0.115 g $NaNO_3$	932	18	–
16	1 ml 40% NaOH solution + 0.0575 g $NaNO_3$	932	18	+
17	1 ml 40% NaOH solution + 0.23 g $NaNO_3$	932	18	–
18	1 ml 40% NaOH solution + 0.2 ml conc. NH_4OH	932	48	+
19	1 ml 40% NaOH solution + 0.2 ml conc. NH_4OH	932	18	+
20	1 ml 40% NaOH solution + 0.2 ml conc. NH_4OH	932	18	–
21	1 ml 40% NaOH solution + 0.126 g $Na_2Cr_2O_7$	932	18	–
22	1 ml 40% NaOH solution + 0.23 g $KMnO_4$	932	18	–

[a] +(Hydrogen embrittlement).
—(No hydrogen embrittlement).

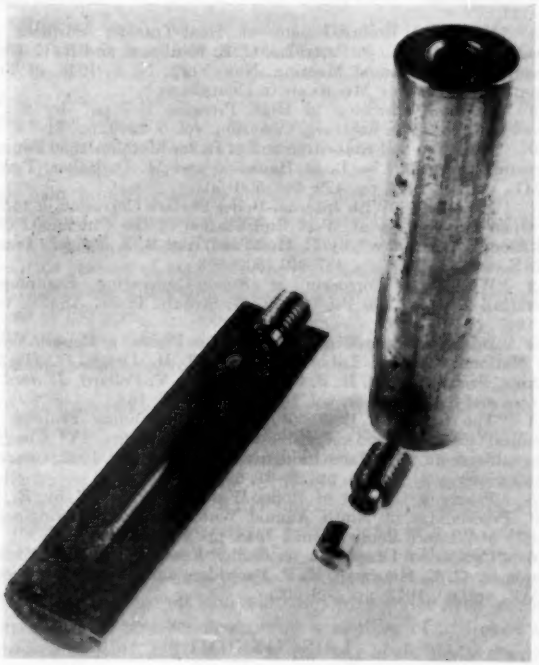

FIG. 12 BOMBS AND PARTS

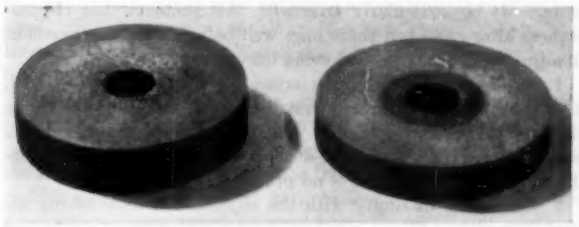

FIG. 13 MACROETCHED SLICES OF BOMBS
(*Left:* Undamaged. *Right:* Hydrogen embrittlement at inner ring.)

added since conditions were supercritical—over 705 F and 3206 psia. With the solutions of NaOH, although critical temperature for pure water was exceeded, a two-phase system at lower pressure was retained because of the presence of the added substance.

It appeared that temperature is, as expected, an important factor in producing damage. Two tests run at 392 F with 40 per cent caustic solution produced no evidence of hydrogen penetration in 18 hr. But at temperatures of 887 and 932 F, hydrogen embrittlement was produced in all cases but one with solutions of 20 and 40 per cent NaOH concentration (13). It will be seen that an elapsed time of only 3 hr produced damage in one instance. The lack of serious attack in test No. 5 cannot be explained on the basis of any obvious evidence.

These bombs were loaded open to the air. Oxygen in the free space of the bomb amounted to about 0.0004 gram, which would oxidize about 0.001 gram Fe to produce 0.0013 gram Fe_3O_4. In test No. 11, 0.1 gram of sodium sulphite was added to a 40 per cent caustic solution. This amount of sulphite was about 25 times as much as would be required to react with the 0.0004 gram of oxygen introduced. This bomb, too, showed attack by hydrogen embrittlement on removal after an 18-hr test at 887 F. Oxygen presumably was used up quickly. Depth of attack, varying from one test to another, appeared to tie in at least qualitatively with length of run, other conditions being equal. In contrast to gaseous oxygen, which is a small fixed amount, the water remaining in the bomb provides a continuing reservoir of bonded oxygen which would permit the extension of the zone of damage.

Since hydrogen embrittlement does depend on the presence of hydrogen, tests were run with oxidizing agents in order to determine if possible reaction with evolved hydrogen would curtail damage. For example, varying amounts of sodium nitrate were added to 40 per cent NaOH solution in preparing charges for tests 12 to 17, run at 932 F. Only with the smallest quantity of nitrate was the damage of hydrogen embrittlement produced.

In the other runs no hydrogen attack was noted even after 72 hr. Similar results were found with sodium dichromate and potassium permanganate.

The final breakdown product of sodium nitrate, if it is participating in the reaction, should be ammonia. Therefore, in order to determine if the reducing property of the nitrate on the one hand, or its decomposition product, ammonia, on the other, was the factor responsible for minimizing attack, tests were run with ammonia added to the caustic solution. In two of the three tests made, hydrogen embrittlement was produced. This seems to indicate that the oxidizing property of the solution provided protection.

The results reported here are indicative but, of course, cannot be applied directly to most situations in power boilers. In most units there will be no necessity for any particular tailoring of the water beyond that usual for normal service since conditions will not be aggravated. In others, a cure involving a decomposable oxidizing agent might well be worse than the disease. For example, the production of large quantities of ammonia from nitrate undoubtedly would be detrimental to copper alloys in other parts of the system.

What to Do About Bonded-Oxygen Attack

Where bonded oxygen is a present or potential threat, engineers should consider all relevant chemical and physical factors. Of these, the broad and basic ones are the local rate of heat input, the local completeness of rinsing, and the character of the boiler water. It is particularly true with this problem that circumstances alter cases but there may well be room for improvement in adjustment along the following lines:

1 Even though dissolved oxygen is not directly involved, avoid the possibility of any contribution to attack by obtaining the best deaeration possible. Use chemical oxygen-scavenging where oxygen remaining after deaeration is appreciable and where decomposition of sulphite is no problem.

2 Maintain alkalinity with the objective of establishing the best over-all conditions for the differing situations of dilute and concentrated boiler water in the particular boiler.

3 Keep refractory baffling and insulation in good repair at critical spots.

4 Where rinsing of surfaces is considered marginal, investigate mechanical aids such as tube spirals or mechanical changes.

5 Chemically clean a boiler which is known or suspected to contain substantial quantities of corrosion products. Removal of accumulations may well provide better local wetting of surfaces and minimize the chance for boiler water to concentrate in crevices between particles.

6 Treat the system so that the condensate and feedwater portions of the circuit will pick up the minimum amounts of iron, copper, and other corrosion products, such as nickel and zinc, from outside the boiler. Neutralizing amines, ammonia, and filming amines all may be valuable in accomplishing this. Although elimination of corrosion products brought into the boiler from outside will not necessarily solve every case of bonded-oxygen attack, it may cut down on severity of damage and effectively reduce unscheduled outages.

Summary

Attack by bonded oxygen has been considered as it applied to specific damage in marine desuperheaters, at a weld in a boiler tube, in a superheater, and by hydrogen embrittlement in a slightly inclined screen tube. Corrosion by the oxygen of water—not that dissolved in water—is the common denominator in these rather different situations.

Laboratory tests produced hydrogen embrittlement in carbon-steel bombs. This attack was minimized or eliminated by the addition of oxidizing agents—although this approach will not be practical in the usual operating boiler.

Careful analysis is required in all cases of bonded-oxygen attack to determine how chemical adjustment of the boiler water or mechanical changes can best be applied to minimize damage.

BIBLIOGRAPHY

1 "A New Approach to the Problem of Conditioning Water for Steam Generation," by R. E. Hall, Trans. ASME, vol. 66, 1944, pp. 457–488.

2 "The Behavior of Highly Concentrated Boiler Water," by C. E. Kaufman, V. M. Marcy, and W. H. Trautman, Proceedings of the Sixth Annual Water Conference, Engineers Society of Western Pennsylvania, 1945, pp. 23–49.

3 "High-Temperature-Steam Corrosion Studies at Detroit," by I. A. Rohrig, R. M. Van Duzer, Jr., and C. H. Fellows, Trans. ASME, vol. 66, 1944, pp. 277–291.

4 "The Corrosion of Alloy Steels by High-Temperature Steam," by G. A. Hawkins, J. T. Agnew, and H. L. Solberg, Trans. ASME, vol. 66, 1944, pp. 291–295.

5 "Attack on Steel in High-Capacity Boilers as a Result of Overheating Due to Steam Blanketing," by E. P. Partridge and R. E. Hall, Trans. ASME, vol. 61, 1939, pp. 597–622; vol. 62, 1940, pp. 711–717.

6 "Concentrating Films; Their Role in Boiler Scale and Corrosion Problems," by H. M. Rivers, Proceedings of the Twelfth Annual Water Conference, Engineers Society of Western Pennsylvania, 1951, pp. 131–145.

7 "Steel, Heat and Water: Localized Formation of Magnetic Iron Oxide in Power Boilers," by H. M. Rivers and W. M. Sonnett, Proceedings of the Midwest Power Conference, vol. 12, 1950, pp. 114–121.

8 "Hydrogen Embrittlement of Heat-Transfer Surfaces in Power Boilers," by E. P. Partridge, C. E. Kaufman, and R. E. Hall, presented at the Annual Meeting, New York, N. Y. 1948, of The American Society of Mechanical Engineers.

9 "Unusual Corrosion in High Pressure Boilers," by S. T. Powell and L. G. Von Lossberg, *Corrosion*, vol. 5, 1949, pp. 71–78.

10 "Corrosion and Embrittlement of Boiler Metal at 1350 Pounds Operating Pressure," by L. E. Hankison and M. D. Baker, Trans. ASME, vol. 69, 1947, pp. 479–486, 500–503.

11 "Experiences With Internal-Boiler Surface Corrosion in 1450-Lb Open-Pass Boilers at West End Station of the Cincinnati Gas and Electric Company," by E. H. Mitsch and B. J. Yeager, Trans. ASME, vol. 69, 1947, pp. 487–491, 500–503.

12 "Wall-Tube Corrosion in Steam-Generating Equipment Operating Around 1300 Psi," by F. G. Straub, Trans. ASME, vol. 69, 1947, pp. 493–503.

13 "Intercrystalline Cracking in Boiler Plates, a Report From the National Physical Laboratory," by C. H. Desch, C. H. M. Jenkins, Frank Adcock, H. J. Gough, and H. V. Pollard, *Journal of the Iron and Steel Institute*, vol. 143, 1941, pp. 93–162.

14 "The Optimum Boiler Water: A Continuing Problem in Chemical Engineering Design," by E. P. Partridge, IV[e] Congres International du Chauffage Industriel, Paris, France, 1952; *Consulting Engineer*, vol. 2, 1953, pp. 28–31, 68, 70, 72.

15 "Precision Control of Boiler-Water Conditions," by R. E. Hall, Proceedings of Ninth Annual Water Conference, Engineers Society of Western Pennsylvania, 1948, pp. 81–95.

16 "Combined Discussion on Boiler Feedwater Studies," discussion by C. E. Kaufman, E. P. Partridge, and R. E. Hall, Trans. ASME, vol. 69, 1947, pp. 500–502.

Discussion

I. B. Dick.[3] In so far as this paper deals with bonded-oxygen corrosion the writer is in agreement. However, questions arise in connection with a few sections of the paper.

It is said that free caustic soda in the dilute water will concentrate and contribute greatly to destruction of the oxide film. This has been said before, and no doubt will be said again. Furthermore, there is a wealth of experimental evidence to support the statement. It would help if someone would explain the chemistry of the reaction between iron oxide and caustic soda and prove it.

[3] Assistant Chief Chemical Engineer, Consolidated Edison Company of New York, Inc., New York, N. Y.

The authors expand the theme by stating that too-little alkalinity also permits bonded-oxygen attack to take place, and indicate that there is a happy medium between too little and too much. This brings up again the old question of whether "captive" alkalinity is much less aggressive than free caustic alkalinity in its participation in bonded-oxygen attack. The writer is of the opinion that it is not, without being able to submit incontrovertible proof.

The authors' illustration of a failed superheater tube, with damage confined to one tube, is not uncommon in power-plant experience. Any bank of uncontrolled parallel circuits is likely to disclose a wide variety of flow rates. This can be detected in a superheater by judicious use of thermocouples and corrected by proper use of restrictors. It would be interesting to have demonstrated that similar damage to overheated superheater tubes would not occur even if no alkalinity were present. Can it not be that water alone is sufficient in an unalloyed or carbon-moly tube at 1100, 1200, or 1300 F?

The authors mention several times in the paper the use of sulphite for oxygen-scavenging. In the writer's experience sulphite has not demonstrated its value for this purpose, and he has yet to be convinced that it is useful for anything but psychological purposes. Since it has now been pretty well demonstrated that sulphite can be decomposed in high-pressure boilers into H_2S and SO_2 which subsequently may damage turbines, condensers, and piping, the question naturally follows, "Why use sulphite anyhow?"

Returning briefly to the main theme of this paper, bonded-oxygen attack, there are several things we can watch for to reduce such attack. Try to keep a vigorous circulation in all tubes in all sections of boilers. Do not permit steam-to-water ratio to get too high. Do not permit sludge accumulations that can form sludge blankets on heat-absorbing surfaces. Keep heat distribution as uniform as possible; eliminate areas of high heat input by proper burner adjustment. Each of these items could be expanded upon considerably, but if all of them can be effected, damage from bonded-oxygen attack will be slight. And this statement stands in the writer's experience even with free NaOH concentrations in boiler water of 1400-psi boilers in excess of 100 ppm.

Finally, the writer wishes to thank the authors for a well-written, thought-provoking paper.

T. J. Finnegan.[4] The authors refer to the direct attack of iron by water as an attack by oxygen "bonded" in the water molecule, H_2O, as distinguished from the oxygen that the water might hold in solution. Another viewpoint might be that it is the attack of iron by water functioning as an acid in which hydrogen ions are displaced by ferrous ions with evolution of the hydrogen as a gas and further reaction of the ferrous ions with the hydroxide ions to produce a solid end product, Fe_3O_4.

It can be shown that iron will react with water even at room temperature to produce measurable quantities of hydrogen and considerable Fe_3O_4. That a large quantity of the metal does not always dissolve is due to influencing factors which hinder the fundamental reaction. As the authors state, the reaction is hindered by the establishment of a protective film on the metal which shields it from the direct attack of the water. This film is undoubtedly an oxide of a composition which depends upon the local conditions.

It would seem then that one of the fundamental principles of corrosion protection would be a knowledge of the factors which would produce and maintain such a protective film as well as the factors which tend to destroy it. At present our knowledge of these topics is largely empirical and research is needed to put it on a sounder chemical basis. Such research is under consideration by the Joint Research Committee at the present time. It is hoped that it will be supported.

W. J. Kinderman.[5] This paper, although specifically directed to the attack of boiler water on carbon steel, also contributes further information related to similar destructive corrosion as experienced, for example, in hot-water pumps and condensate-drainage devices. It is a matter of common observation that black oxide is formed on the inside surfaces of such mild applications as low-pressure hot-water heating lines where the coating itself generally affords protection which arrests further attack. In conjunction with higher pressures and velocities, notably jet impingement, however, the oxide is subject to continuous removal and progressive attack may then occur with resulting destruction of carbon-steel parts.

It has been variously reported that chromium-bearing steels even in small percentages (4130 steel, for example) resist the progressive deterioration noted for carbon steels under similar conditions of exposure. Comparative tests conducted in the Research Laboratory of the writer's company with near steam temperature condensate at 250 psi initial pressure flowing at velocities destructive to carbon steel serve to confirm the beneficial effect of chromium. Stainless steels showed every evidence of providing full protection against this type of attack.

Since the paper cites a failure of type 347 stainless steel but implies that the mechanism of failure differs from that of carbon steel, any additional available data which would help to distinguish the respective failures should prove to be highly enlightening.

It would seem that erosion and possibly cavitation effects could play an important part in the destructive action noted quite apart from that attributable to pure chemical attack. In the case of the desuperheater tube failure is it not likely that the violent agitation associated with boiling water could provide a continuous scouring which would free the surface of corrosion products and thus favor a progressive attack? Any supplementary comments which the authors may feel justified in advancing would serve to extend the benefits of this paper.

The authors not only deserve to be complimented on their contribution to a better understanding of the boiler-corrosion problem but also for its application to the related problems of reference.

A. R. Mumford.[6] In the presentation of this paper reference was made to "high" heat transfer. The writer would like to know what rates are high.

The ASME Special Research Committee on Furnace Performance Factors has measured and reported many measurements of rates of heat transfer in many furnaces. The highest of these, recalled by the writer, exceeded 200,000 Btu per sq ft per hr. Is that high? The furnace in which these measurements were made was fired by natural-gas fuel,[7] and the points at which this magnitude of heat transfer was noted were in the vertical tubes of the rear wall and the low slope of the roof. The former were about in line with at least one of the burners and the direct impingement of hot gases was noted. Mixing of air and fuel was excellent so that combustion was completed,

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[5] Director of Research, Yarnall-Waring Company, Philadelphia, Pa. Mem. ASME.

[6] Research Department, Combustion Engineering, Inc., New York, N. Y. Fellow ASME.

[7] "Variation in Heat Absorption in a Natural-Gas-Fired, Water-Cooled Steam-Boiler Furnace," by A. R. Mumford and R. C. Corey, Trans. ASME, vol. 74, 1952, pp. 1191–1215.

according to visual observations, less than 10 ft from the burner wall. The rear wall was about 20 ft from the burner wall. Temperatures in the gas stream less than 10 ft from the burner wall were high enough to melt platinum and platinum 10 per cent rhodium. Near the rear wall gas temperatures were indicated to be about 3000 F. The measurements were made after 8 or 9 years of operation and about 4 years have elapsed since. No tube trouble has ever been reported on either the rear wall or the roof. The writer personally examined all of the furnace walls from the furnace side and found no indication of overheating. History, therefore, indicates that heat-absorption rates of over 200,000 Btu per sq ft per hr were not too high in this installation.

In order to be helpful to designers of heat-absorbing equipment it would be advisable to stipulate what rate of transfer is too high rather than that it is high.

Authors' Closure

The authors are in agreement with Mr. Dick that the chemistry of the intermediates in the reaction between iron oxide and caustic-soda solution is obscure. Clarification in this area would certainly be welcome, although it is problematical whether additional knowledge will revolutionize the approach to solution of the problem.

In connection with Mr. Dick's questioning of whether captive alkalinity is much less aggressive than free-caustic alkalinity, there is an interesting side effect which sometimes comes into play. With appreciable free alkalinity tubes fail from gouging. When free caustic is reduced or eliminated, a tendency is noted for siliceous materials to form a scale on the tubes. Here again the end result may be failure, arising this time from overheating.

With respect to the failed superheater tube, we agree that water or rather steam alone can oxidize steel or low alloys. In fact, the case cited was apparently largely of this nature.

We agree and disagree with Mr. Dick's viewpoint on sulphite for oxygen scavenging. In tight high-pressure boiler systems with negligible make-up, units which are common in the utility field, sulphite may be of negligible benefit. However, with high-pressure industrial boilers using considerable amounts of make-up, we feel that sulphite can have value in removing the last traces of oxygen.

We certainly agree with Mr. Dick that if design and operation permit the adjustment of many relevant conditions, primarily mechanical in nature, danger from bonded oxygen attack should be at a minimum.

We appreciate Mr. Finnegan's comments on alternative mechanisms which might be associated with the water-iron reaction. Fundamental studies leading to a better knowledge of protective films and their maintenance certainly are desirable.

Mr. Kinderman asked for supplementary information on the failure of the stainless-steel desuperheater section. Cavitation and erosion may have played a part here in initiating failure at certain points but wastage was really at a minimum, indicating that thermal shock and repeated stress were perhaps of predominant importance.

To Mr. Mumford's query on how high is high with respect to heat transfer, the honest answer is that we do not know. Perhaps what we should have said is that "high" heat transfer can be contributory to difficulty, particularly where other factors such as inadequate rinsing, deposit formation, or the like, exist. The term speeding applied to automobile traffic may be analogous. In general, high speed contributes to accidents but the character of the road can make 70 miles an hour speeding at one place and only 35 miles an hour speeding at another.

We thank the discussers for their comments.

Corrosion of Steel in Boilers—Attack by Dissolved Oxygen

By H. A. GRABOWSKI,[1] NEW YORK, N. Y.

The role of oxygen in the corrosion of internal surfaces of boiler-furnace-wall tubes has been discussed widely without arriving at consistent conclusions in respect to the maximum concentration of oxygen that can be tolerated and the exact cause of the initial pit formation at the tube surface. The present paper reviews the various theories that have been proposed to explain the severe, although local, corrosion of furnace-wall tubes in high-pressure boilers. Results of field tests of the oxygen record in utility-type boilers are presented. Metallurgical significance of failures and welding techniques also are discussed.

Introduction

ONE of the most widely discussed topics in power-plant chemistry is the exact role of oxygen in the mechanism of corrosion in steam boilers. That a variance of opinion exists among known authorities in the field is proof that sufficient fundamental data, as acquired in laboratory investigations, is lacking to corroborate opinions formed from numerous field observations.

The fundamental information concerning the reaction rate of boiler steel with water and steam, over a range of temperatures and pressures, unfortunately is limited and there is urgent need to initiate basic investigations to obtain the necessary data. Since oxygen is believed to have a significant role in the reactions of water and steel, it is quite evident that controversy concerning its importance is a natural development. To supply the missing data, the Sub-Committee on Corrosion, of the Joint Research Council on Boiler Feedwater Studies, is organizing a research program to study the variables affecting the corrosion of steel in contact with water.

The purpose of this paper is to review one phase of the problem, that of oxygen corrosion in boilers, and to indicate why it has not been resolved.

Numerous papers have been presented before technical societies bearing upon the cause and occurrence of corrosion; but the data presented have been obtained generally from the following:

1 Low-temperature operation extrapolated to the condition as found in the field.

2 Material accumulated from experience which appeared to be applicable to the case at hand but did not necessarily present a clarification of a similar problem at another location.

3 Limited laboratory-test results combined with field observations to answer specific boiler-tube-corrosion problems.

The present paper will deal more specifically with tube corrosion as found in boilers operating above 800 psi. Generally, greater emphasis is placed on the importance of conditioning water supplied to boilers at the higher pressures, with the result that scale and sludge problems, common at the lower pressures, are encountered rarely. At higher pressures, the principal problems experienced are those associated with corrosion and silica. The basic considerations are the factors which control the corrosion rate of boiler steel. To date, information on this problem usually has involved personal interpretations of the difficulties; hence it is not surprising that controversy as to the validity of the conclusions would result.

The Corrosion Problem

Although the number of boilers that have experienced severe corrosion are relatively few, the extent and intensity of attack in the affected units has attracted attention. Probably the most important single obstacle to attainment of a solution is in the disagreement among parties concerned as to the cause.

Corrosion attack has been observed in (*a*) vertical furnace-wall tubes; (*b*) at, near, and away from shop-welded areas; (*c*) at and near field welds containing backing rings; (*d*) at and near field welds without backing rings; (*e*) near welded areas in which oxyacetylene and electric welding techniques have been used; (*f*) at points of discontinuity between two welded fins on a furnace-wall tube; (*g*) in locations where a tube portion was mechanically stressed in rolling into a header; and (*h*) in areas of both high and low heat transfer in the furnace.

Oxygen has been suggested as one of the primary causes of such attack. The principal point of dispute in the mechanism of corrosion is the role which oxygen assumes in the attack of boiler-tube metal.

The corroded area generally is characterized by the presence of a black, dense, and brittle layer of oxide overlaying the point of wasted tube metal. The tube metal in the pitted section is usually brittle, contains intercrystalline cracks, and shows evidence of decarburization in the embrittled area. Examples of this condition will be cited later.

In contradiction to the theory of oxygen influence on this type of tube wastage, other factors have been proposed as the cause of tube attack. These include:

1 Excessive local heat-transfer rate.
2 Inadequate circulation of water.
3 The presence of backing rings.
4 The occurrence of free hydroxide under which conditions a protective oxide cannot form over the welded area.
5 Faulty metallurgical structure.

Were such conditions related to mechanical design and fuel-burning problems, it would be reasonable to assume that a change in boiler design would provide a solution. However, available data, as well as the author's experience, indicate that changes in mechanical design have not been necessary in preventing recurrence of such attack. In nearly every instance, chemical changes in the boiler water and improvements in the feedwater system, such as deaeration, have resulted in control of even the most severe corrosive activity.

If the problem were affected consistently by factors of typical

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[1] Research Engineer, Combustion Engineering, Inc. Mem. ASME.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, November 5, 1953. Paper No. 53—A-226.

power-plant operation, the solution and the mechanism might be deduced readily. However, the controversial metal attack has been observed in power stations where (*a*) boiler-water alkalinity has been maintained by essentially a co-ordinate phosphate control; (*b*) where the pH of the boiler water has been adjusted with caustic to a value somewhat under 10; (*c*) where the pH value was maintained by caustic at a pH near 11; and (*d*) where no chemicals are added to the boiler. From the standpoint of pH value alone, if heat transfer and fluid flow were the basic causes for the corrosive action it would appear that boilers having no chemical treatment and a pH value of about 8.5 would be the most seriously affected. This condition has not been observed in the field. Thus it would appear that the maintenance of an alkalinity residual alone does not determine whether adequate protection is afforded.

Such tube-metal attack has been found in boilers having welds without backing rings, at welds where backing rings have been used, and in boilers where both welding techniques have been utilized. For practical consideration, it can be said that backing rings alone are not the basic cause of this form of attack. Since the corrosion has been observed distant from a weld, in a straight run of tubing, it would follow that the metal wastage must involve other factors than welding technique in initiating the break in the protective surface of the metal.

Backing rings or welding reinforcements are mentioned frequently as being instrumental in promoting the initial attack of the tube metal. Proponents of this theory envision a disturbance in the flow area, due to the reinforcement, which favors the trapping of a layer of steam and the deposition of iron oxides and other solids in the vicinity of the ring. A slight temperature rise which results from dead spaces at the sides of the reinforcement favors additional deposition and accelerates oxidation. Corrosion of the type under consideration has been the exception rather than the rule. The joint designs and welding procedure under discussion have been used for many years and the techniques developed are standard. It is reasonable to conclude that the corrosion problem would be universal and encountered in numerous installations if the welding reinforcement were the basic cause of attack of the metal surface. Since most boilers are free of such difficulty, and only a few are so affected, it may be concluded that welding design and technique may be eliminated as prime factors.

Although the method of the control of pH of the boiler water varied in the cases where tube corrosion was encountered, it was observed that without exception a number of factors were evident in each instance. Plant and test records revealed the presence of larger quantities of oxygen and preboiler corrosion products in the feedwater. In practically every case a reducing agent, for oxygen scavenging, was not utilized in treating the boiler or feedwater. Poor deaeration during normal operating conditions, at low steaming loads, and during start-up and shutdown of a boiler were noted in each instance.

History and Field Experience

Professor Straub[2] cited two examples of the type of attack which he attributed to the presence of oxygen in the feedwater in the absence of a suitable oxygen scavenger in the boiler water. The failures he described were brittle in nature, contained intercrystalline cracks, and had become decarburized in the embrittled section. Corey,[3] in his review of the two cases of failure, showed that in one, fluid velocity and temperature surveys had disclosed a normal condition existing in the affected sections. Failures at both stations were characterized by deep internal pits on the fireside of the tubes which were filled with dense plugs of Fe_3O_4. Corey indicates that, from his observations, the tube damage at both stations was initiated by small spots of rust which formed in the tubes during times when high concentrations of oxygen were in contact with the metal, such as after drainage or after filling a boiler with poorly deaerated and untreated feedwater. In the absence of experimental evidence, this hypothesis, in the author's opinion, offers the most logical and practical approach to the corrosion problem under discussion.

Corey[4] in his excellent review to the ASTM (1948) further commented on one of the foregoing failures, stating, "Although there was appreciable copper in the deposits of Fe_3O_4 associated with the damaged tubes, all the facts in the case suggested that the pitting was caused by oxygen, either during outages or during operation, and the pits became foci of rapid attack during protracted periods of operation at full load."

If one is to avoid the conditions necessary to cause initial pitting, it would be logical to minimize the oxygen content in the water supplied to the boiler. One may inquire with reason, as to the concentration of oxygen and the maximum quantity that can be tolerated, or as to the oxygen history in a high-pressure station over the period of its operation. Generally, the answers supplied to the author on these points were values corresponding to guarantee specifications. Upon checking a series of stations the following were found:

1 Many stations tested for oxygen only a few times a week and these checks were made during normal conditions of operation.

2 On starting up a boiler, most units were filled with undeaerated condensate. A majority of the stations treated the water as it was filled into the boiler.

3 For a period of 2 to 4 hr after the start-up of a boiler, oxygen contents greater than 0.5 ppm were found in the feedwater added to the boiler.

4 In many stations, at low loads, an appreciable leakage of air into the low-pressure heaters was observed. The drips were highly contaminated, containing up to 0.5 ppm of oxygen. In stations where all deaeration was effected in the condenser and the drips were returned to the feedwater system, a significant increase in oxygen to the boiler was observed.

5 In several stations where all deaeration is completed in the condenser, and surge tanks float on the condensate line, an appreciable pickup of oxygen was observed when water entered the feed system from the surge tanks.

In a majority of the stations, under normal conditions, the oxygen content in the feedwater added to the boiler is between 0.005 and 0.02 ppm. A few stations operate with oxygen contents between 0.02 and 0.04 ppm. Where a chemical agent for oxygen reduction is employed it would appear that, under normal operation, no metal attack should be anticipated.

Although the various theories of the cause of the corrosion can be argued at length, it cannot be overlooked that in the stations known to the author, where severe pitting was observed, oxygen in significant quantities was present in the boiler feedwater. A few illustrations may serve to indicate the scope and recurrence of the oxygen problem.

Case A. Boiler 1 in Station A, operating at 1250 psi, was in service without encountering tube problems for nearly 10 years. Boiler 2 was placed in operation about 2 years prior to the date

[2] "Wall-Tube Corrosion in Steam-Generating Equipment Operating Around 1300 Psi," by F. G. Straub, Trans. ASME, vol 69, 1947, pp. 493–499.

[3] "Internal Corrosion of Furnace Tubes of High Pressure Boilers," by R. C. Corey, *Combustion*, March, 1947.

[4] "Corrosion of High-Pressure Steam Generators: Status of Our Knowledge of the Effect of Copper and Iron-Oxide Deposits in Steam Generating Tubes," by R. C. Corey, Proceedings of the ASTM, vol. 48, 1948, pp. 907–942.

when the corrosion problem was first encountered. Deaeration for both units was completed in the condenser. When the second boiler was placed in service, the following changes in the plant cycle were put into effect: The boilers were operated at a reduced load between midnight and 6 a.m. instead of base-loaded. A portion of the high-pressure steam was desuperheated and eventually exhausted into an existing low-pressure boiler system. Condensate taken from the deaerator serving the low-pressure boilers was transferred to the high-pressure-boiler feedwater system. Owing to the heavier make-up-water requirement, the evaporators in service frequently carried over solids into the feedwater system.

An oxygen scavenger, added at the inlet of the boiler feed pump from a pot-type feeder and proportioned by the load change across the pump, proved to be effective during the period when only one boiler was in operation. Since the addition of the chemical selected for scavenging purpose could not be checked effectively by chemical test, its benefit to the system was of doubtful value when high quantities of oxygen, over short intervals, were present in the feedwater. Actually, the smallest quantities of scavenging agent were added during the periods when the oxygen content was the highest.

Fig. 1 shows the increased oxygen content in the water supplied to the boiler during low-load operation. One of the low-pressure heaters, normally operating under pressure, dropped to vacuum during the low-load operation. The drips from this heater, which were added directly to the feedwater system, contained considerable oxygen, up to 0.5 ppm, when the steaming rates were low. During normal operation, when the heater pressure was above atmospheric, the oxygen content in the drips was less than 0.1 ppm.

Fig. 2 illustrates the increase in the oxygen content of the feedwater resulting from the introduction of a poorly deaerated condensate from the low-pressure-boiler system to the high-pressure

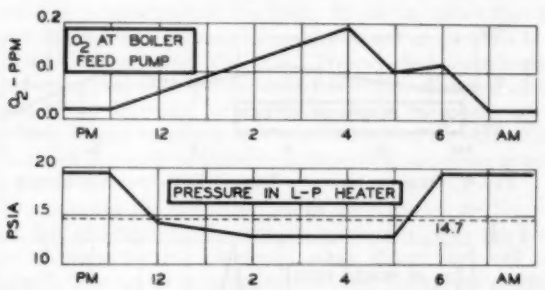

Fig. 1 Oxygen Increase Resulting From Air Leakage in L-P Heaters

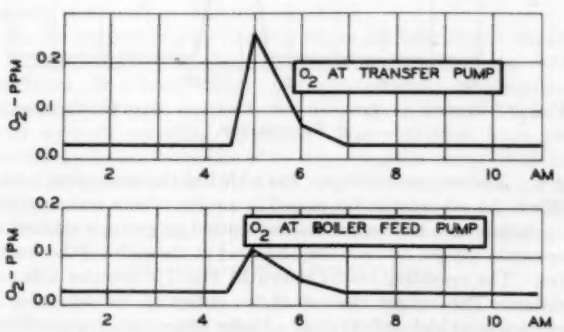

Fig. 2 Oxygen Increase Resulting From Addition of Undearated Condensate

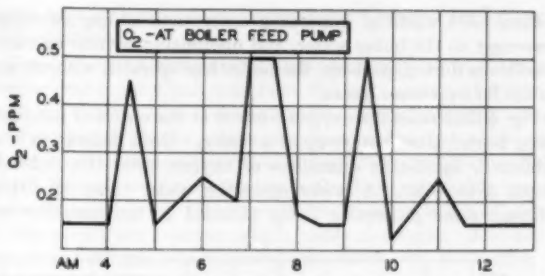

Fig. 3 Oxygen Variation During Boiler Banking

units. Normally, the low-pressure boilers were banked during the night operation. When they were placed in service, the deaerating heater supplying water to both the low and high-pressure systems was overloaded and a poor product was obtained for about 3 hr. This condition occurred daily and oxygen, in substantial quantities, was introduced into the high-pressure boilers.

In this system a surge tank floats on the suction side of the boiler feed pump. In operation, the surge tank supplies the condensate to the feedwater system whenever unusual operating conditions warrant. Fig. 3 represents a period of operation when one boiler was being banked and the second was in operation. A common feedwater supply served both.

The increases in oxygen content mentioned occurred mostly during night operation when no samples were taken for control analysis. When a chemical analysis was performed by the plant chemist, during the day shift, a low residual of oxygen was observed. It is obvious that in evaluating the effect of the quantity of oxygen in relation to the corrosion problem, it is necessary to determine the frequency of any operating condition that introduces oxygen into the cycle. A continuous record offers the only positive approach to a determination of the history of oxygen control in any particular station.

Failures in the high-pressure boilers occurred at welded areas, at a point away from any welds in straight runs of boiler tubing, in areas where the tubes were stressed by bending in rolling them into the header, and at points of discontinuity between two welded fins. In all these cases the principal corrosion was observed in a portion of the furnace where the heat-transfer rate was relatively low.

Although other factors, such as alkalinity, may have influenced the corrosion process, it is more significant that the metal attack occurred only when oxygen in excessive quantities entered the feedwater system and addition of the oxygen-scavenging agent was poorly controlled.

Case B. Similar failures occurred at another station in a boiler operating at 950 psi, after nearly five years of service. The principal metal attack occurred at or near shop welds. Some of the corrosion occurred at sections away from welded areas. Since the principal metal wastage was in a relatively high-input area, the logical conclusions as to the cause appeared to be local high heat input and disturbances to flow by welding reinforcements at the weld junction. Metallurgical analyses and tube-metal-temperature surveys clearly pointed out that overheating had not occurred in the affected area and there was no evidence of excessive heat input.

An oxygen scavenger was not added to the boiler water. At each outage, large quantities of iron oxides and copper were found. The accumulation of deposits was attributed first to corrosion of metal within the boiler. The results of a survey subsequent to these failures conclusively showed that the principal source of the oxides was the preboiler surfaces which came in contact with a lower-pH feedwater.

Since acid washing, improving deaeration, adding an oxygen scavenger to the boiler water, and minimizing adverse operating conditions during start-up, the boiler has operated without such failure for over seven years.

Fig. 4 illustrates the oxygen content at the outlet of the deaerating heater after a start-up of a boiler. Until deaeration is established, significant quantities of oxygen enter the boiler for about 3 to 4 hr. A boiler operating under these conditions, without some protection being afforded to the metal, over a

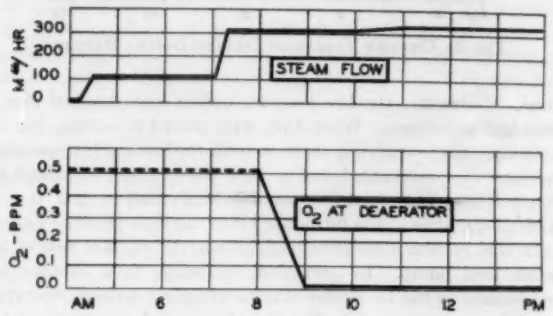

Fig. 4 Oxygen in Boiler Feedwater During Start-Up Operation

period of years, can develop a corrosion pit in a weakened area which is capable of promoting accelerated metal attack.

Case C. A boiler operating at 900 psi encountered tube failures resulting from corrosion at welded areas and in straight sections of furnace-wall tubing after nearly six years of operation. An oxygen scavenger was not added to the boiler feedwater. The boiler water pH was considerably lower than in Cases *A* and *B*, in general, less than 9.0. Appreciable quantities of iron oxides and copper were formed by the low-pH feedwater.

A deaerating heater was employed to maintain low quantities of oxygen in the feedwater. As in Case *B*, the oxygen content upon starting up was high for about 3 to 4 hr, but under normal conditions the oxygen residual in the feedwater was relatively low, being about 0.01 ppm.

When the failures were noted, a survey of the furnace-metal temperatures showed normal distribution without any excessive temperature being observed. The application of corrective measures to minimize the introduction of oxygen into the boiler, the addition of sodium sulphite to the boiler feedwater, and an increase in feedwater pH by removal of carbon dioxide from the evaporated vapor make-up to the system, prevented further tube failures. The boiler has now operated seven years without a recurrence of the corrosion problem.

Oxygen is commonly introduced into the feedwater cycle in systems employing deaerating heaters or deaerating condensers during starting, operation at low load, faulty evaporator operation, with feedwater heaters improperly vented, or where air leakage into the turbine seals is excessive. The following examples illustrate incidents in operation which result in the introduction of oxygen into the feedwater:

1 The load on a high-output high-pressure boiler was reduced gradually to about 25 per cent of rated capacity and then was brought back to normal. During this time, the deaerating-heater pressure, normally above atmospheric pressure, dropped to a few inches of vacuum. An increase in the oxygen content of the boiler feedwater was observed during the low-load period, the major rise occurring during the interval when the steam load was brought up rapidly. The variation in oxygen content of the water is shown in Fig. 5.

2 In another high-capacity boiler operating at a high pressure, an increase in the oxygen content was observed each night when the load dropped below 40 per cent of rated capacity. At normal loads, the oxygen in the feedwater from the deaerator was about 0.005 ppm. During the low-load condition, the oxygen content rose to about 0.015 ppm. The variation in oxygen content is shown in Fig. 6.

3 In a third station, the feedwater from the deaerating heater to a 1400-psi boiler normally contained about 0.008 ppm of oxy-

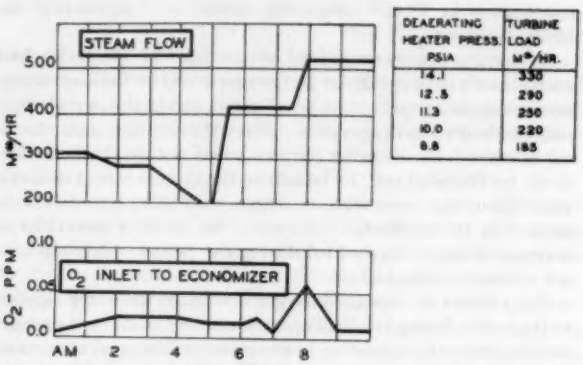

Fig. 5 Effect of Drop of Pressure in Deaerator to Vacuum

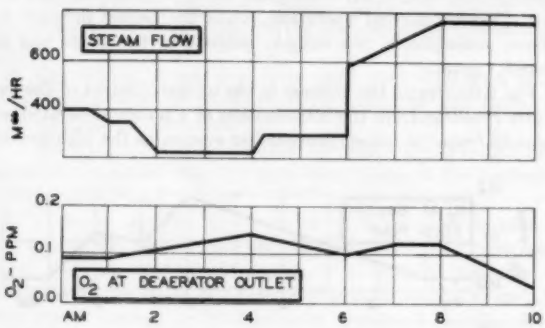

Fig. 6 Oxygen Increase During Low-Load Operation

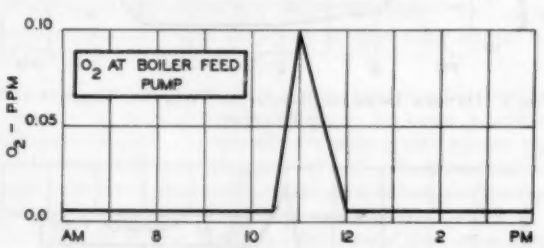

Fig. 7 Effect of Insufficient Venting Before Cutting in Evaporator

gen. The evaporator vapor was added at the deaerating heater. When the evaporator was placed in service, after a scale-cracking operation, and the vapor was not vented properly, a marked increase in oxygen content was observed at the outlet of the deaerator. The operating record shown in Fig. 7 illustrates this condition. The oxygen content at the outlet of the deaerator increased to as high as 0.09 ppm. Under these operating conditions poorly deaerated water was added to the boiler over a period of 1.5 hr.

4 In a fourth station, operation of the deaerating heater was impaired by the introduction of a large quantity of water from a surge tank into the condensate line. The increase is shown in Fig. 8. During this time, the oxygen content of the water supplied to the deaerator increased to values as high as 0.5 ppm. Under normal operating conditions, the effluent from the deaerator

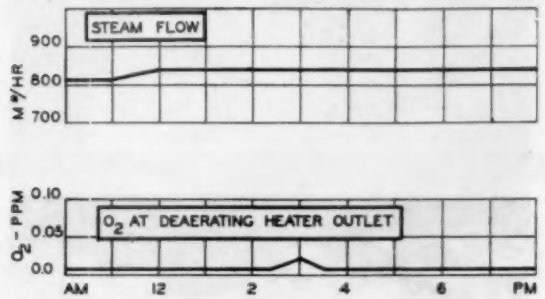

FIG. 8 EFFECT OF LARGE ADDITION OF SURGE WATER TO DEAERATOR

contained about 0.005 ppm of oxygen, but during the surge-water addition, the oxygen content in the water from the deaerator increased to 0.03 ppm.

The operating conditions described, normally occurred during periods of the day when laboratory personnel were not available to collect and analyze samples of water. To the author's knowledge, oxygen determinations rarely, if ever, are conducted during the off-peak, night, or week-end low-load operation. When a record of oxygen appears in the plant data, it most frequently represents analyses conducted at normal conditions.

In the author's opinion, it is not possible to evaluate the critical content of oxygen or its role in the mechanism of corrosion from data accumulated in the field. It can be stated that the boiler which receives almost oxygen-free water at all times is an exception rather than the rule. Present deaerating-heating standards guarantee not over about 0.005 ppm of oxygen at the outlet of the deaerator. These requirements have been set on the basis that a reduction to the minimums of an undesirable component is certain to improve an otherwise hazardous operating condition.

In evaluating the role of oxygen in the corrosion mechanism, from field observations, the problem is complicated by the presence of other factors. Generally, when water conditions are changed, steps are taken to improve not only the oxygen problem but all the other factors as well. In retrospect, it is indicated that the oxygen problem was basically the underlying cause of the metal attack. Similarly, the other factors can be stressed equally as important causes of the attack.

In the analysis of case histories where the tube attack results in brittle failures, and where tube-metal temperatures and fluid velocities are normal, oxygen has been observed in quantities exceeding 0.05 ppm in the boiler feedwater. This indicates a poor over-all operating condition. Where failures have occurred, a solution was achieved by improving the operation of the preboiler apparatus and by modifying the chemical control of the feed and boiler waters. Mechanical changes for improving fluid velocities and heat distribution have not been necessary.

METALLOGRAPHIC OBSERVATIONS

A number of authors have shown the metallurgical structure of the tube metal in the vicinity of the failures to be associated to some degree with oxygen attack. The metal in the region of the failure is generally brittle. Popularly, the metal in the affected area is referred to as having been weakened by hydrogen embrittlement.

The embrittled structure is also commonly ascribed to excessive heat input which may raise the temperature of a film at the tube surface. This temperature may be high enough to cause a concentration of boiler-water components, but not necessarily high enough to cause a structural change in the outer area of the tube. The caustic concentration in the film presumably breaks down the protective oxide layer and this action is accompanied by the steel-water reaction which yields hydrogen. The latter component diffuses into the metal and reacts with the carbon to form methane gas. The methane molecule does not readily diffuse but concentrates at the grain boundary and exerts a force which tends to pry the grains apart.

This approach, in the light of limited experimental data, offers an interpretation of the corrosion taking place where an embrittling action is observed without any indication of a change in structure, due to overheating, being noted in the metal outside the corroded portion. In this mechanism only a small rise in temperature is required to initiate the corrosive action. Although this explanation has merit, it also must assume that, as long as the same heat flux exists in a furnace, the problem can be alleviated only by minimizing the effect of the factors in the concentrating film, as caustic. Thus, in effect, relief is attained by dilution of the chemicals in the film by the addition of compounds such as chlorides.

Equally severe corrosion has been observed in boilers where no free hydroxide is maintained, the alkalinity being maintained by the co-ordinated phosphate treatment, or where no chemicals are added to the boiler water. Temperature surveys have not shown values other than those anticipated for the operating pressure.

It is easy to see why a controversy exists in explaining the nature of the corrosion. It is apparent that although the concentrating-film theory has merit in explaining the corrosive action, other factors play a more significant role in causing the breakdown of the metal surface.

The brittle-type failures have occurred at various locations in the furnace. A review of several of the specimens will indicate the severe nature of the attack:

1 Fig. 9 shows the corrosion occurring in a section of tube which was stressed mechanically while rolling into a header near the top of a furnace. As can be seen, the deposit is hard and dense. Fig. 10 shows the deep-etched appearance of the tube metal at the corroded area. An embrittled layer is visible. Fig. 11 illustrates at ×100, unetched and nital etched, the area in the dark zone of Fig. 10. Decarburization and intergranular cracking are clearly evident. Fig. 12 illustrates at ×100 the unaffected area in Fig. 10. The structure of this metal is normal.

The tube metal in Fig. 10 was studied to determine the nature of materials observed in the voids at the grain boundaries. Also, the metal structure was observed at various depths from the inside to the outside surface of the tube specimen in the vicinity of the failure. Fig. 13 shows the inside surface of the corroded area at ×100. Fig. 14 shows the metal layer 0.064 in. from the inside surface at ×1000. Fig. 15 shows layers at 0.115 in. and 0.158 in. from the inside surface at ×1000. The 0.115-in. layer was made at the boundary line between the dark etched and light layers in Fig. 10.

Figs. 13 and 14 show a decarburized section with intergranular cracks. At the 0.115-in. layer in Fig. 15 a trend to normal steel is evident. This narrow zone exhibits spheroidized carbides which tend to agglomerate at the grain boundaries. The outermost layer shows a normal tube structure.

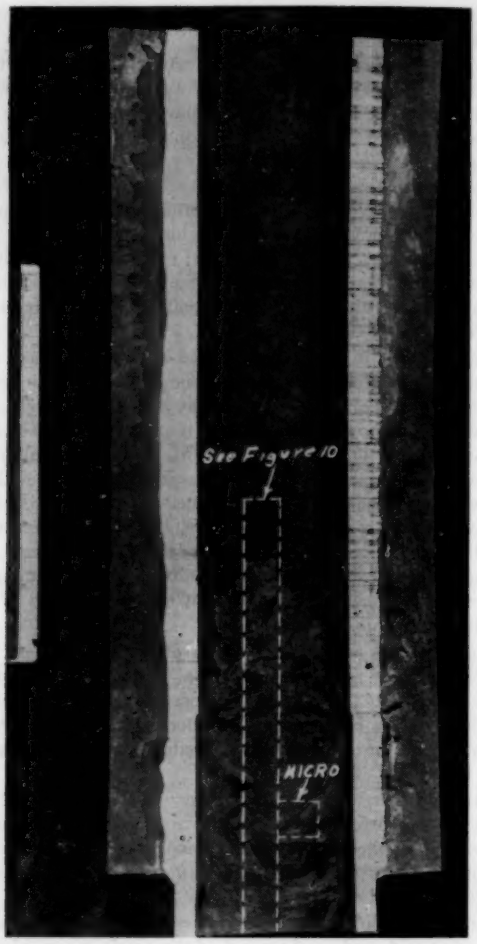

FIG. 9 CORROSION AT TUBE BEND, INSIDE SURFACE MACHINED

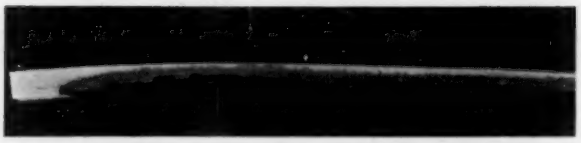

FIG. 10 EMBRITTLED CORROSION AREA SHOWN IN FIG. 9

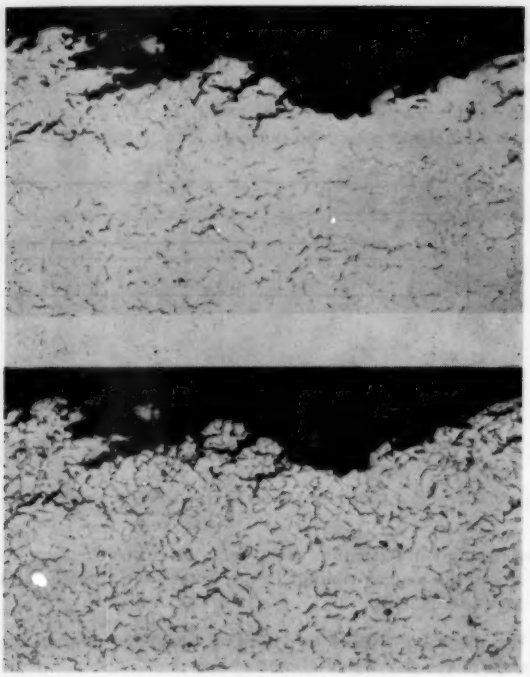

FIG. 11 MICRO OF DARK AREA IN FIG. 10; × 100
(*Top*, unetched; *bottom*, nital etched.)

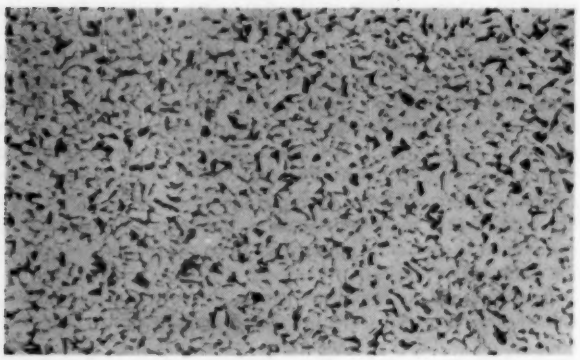

FIG. 12 MICRO OF AREA ETCHING LIGHT IN FIG. 10; ×100—NITAL ETCHED

Microscopic examination of the decarburized zones showed the presence of a number of grain-boundary inclusions. The etching techniques used indicated that the inclusions may consist of iron oxides. An investigation is under way to determine the composition of the inclusions. Should the results of the preliminary etching tests be confirmed, the term hydrogen embrittlement may be a misnomer in that oxygen also takes part in the reaction.

It has been stated that to blame oxygen for problems associated with hydrogen embrittlement leads to an absurd set of conclusions, since the action of oxygen is to unite with hydrogen evolved at the cathode of an electrolytic cell and not with the iron. In view of the limited experimental evidence, and the lack of agreement of popular theory and actual observations, it would appear to be more logical to pursue thorough fundamental investigations instead of amplifying data gathered, in most instances, at room temperature.

2 Fig. 16 shows corrosion in the area of an oxyacetylene weld with a backing ring. Fig. 17 shows an embrittled section in the vicinity of the corroded area. Metallographs made of the area of corrosion and of the weld metal showed that the attack on the weld metal was slight in respect to the adjacent tube metal.

3 Fig. 18 shows a brittle failure in a tube section of a boiler operating at 1250 psi. Note that the attack occurred at a distance from a flash weld. The tube area at the failure was embrittled as shown in Fig. 19.

4 Fig. 20 shows a brittle failure near a shop flash weld. The area at the point of failure was embrittled. In the dark etched area, the steel was decarburized with evidence of intergranular cracking.

5 Fig. 21 shows the inside surface of a tube after an oxyacetylene pass was made on the outer surface. Prior to completing the pass, the inside surface was buffed with a wire brush. Thus,

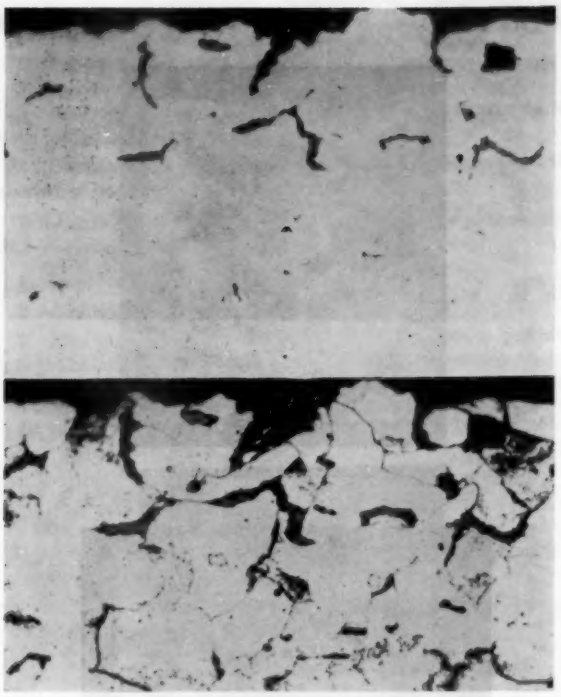

FIG. 13 MICRO AT INSIDE CORROSION SURFACE OF FIG. 9; ×1000
(*Top*, unetched; *bottom*, nital etched.)

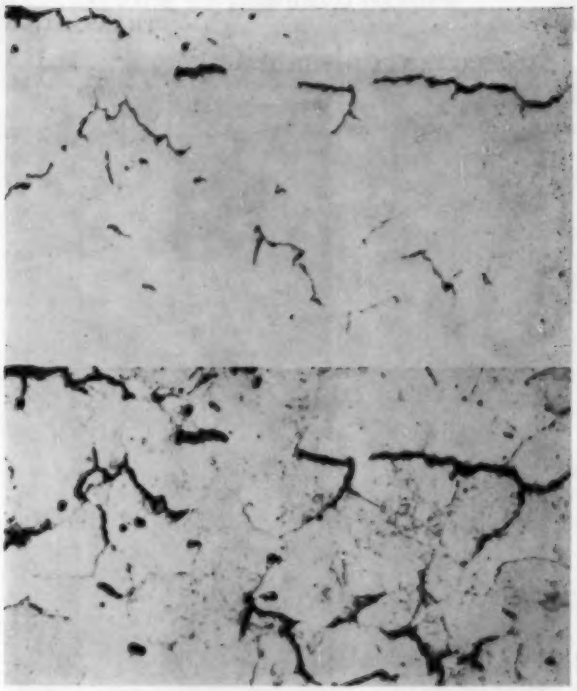

FIG. 14 MICRO 0.064 IN. FROM INSIDE SURFACE OF FIG. 13; ×1000
(*Top*, unetched; *bottom*, nital etched.)

FIG. 15 MICRO FROM INSIDE SURFACE OF FIG. 13, NITAL ETCHED; ×1000
(*Top*, 0.115 in. from inside surface; *bottom*, 0.158 in. from inside surface.)

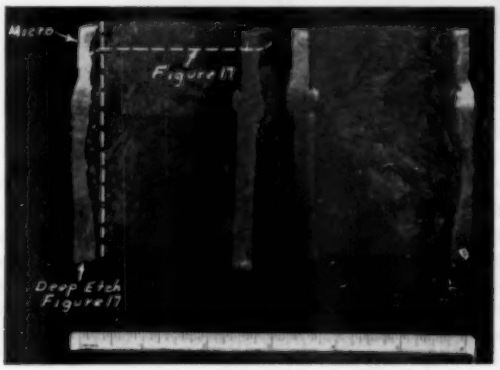

FIG. 16 CORROSION AT OXYACETYLENE FIELD WELD WITH BACKING RING

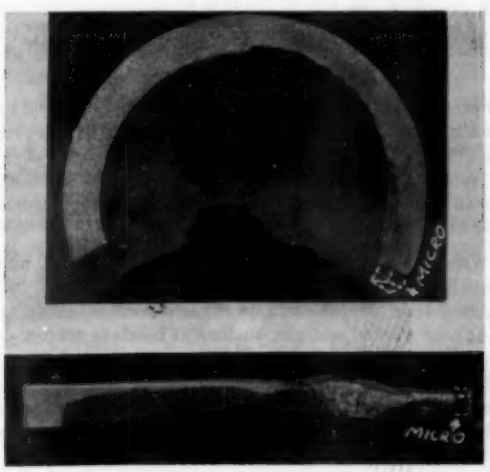

FIG. 17 EMBRITTLED AREA AT WELD, FIG. 16

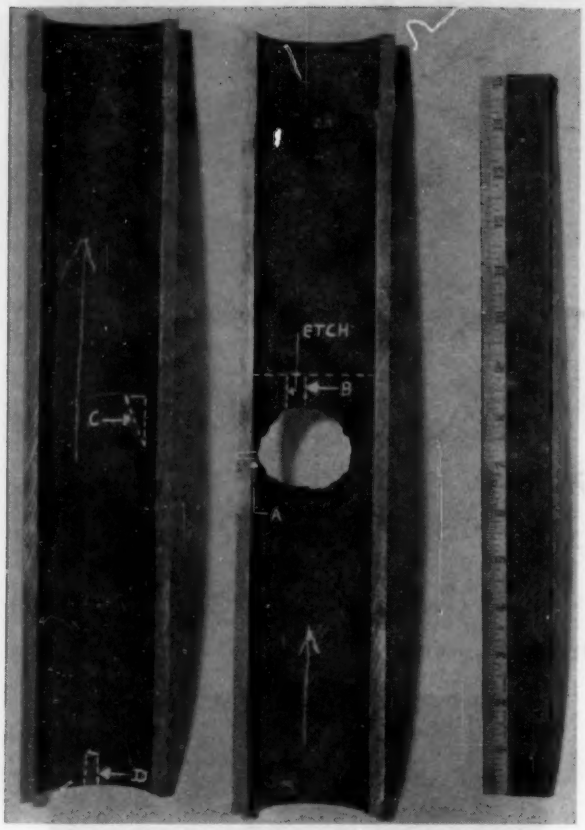

FIG. 18 CORRODED AREA AWAY FROM SHOP WELD

without the presence of mill scale, a fluffy oxide layer was obtained under the location where the welding pass was completed. Fig. 22 shows the appearance of the tube after making an oxyacetylene pass on a tube specimen which was cleaned with a mixture of muriatic and nitric acids. The disturbance on the tube surface is seen clearly. From these tests it could be indicated that a focal point for attack can be set up by welding, but the metal deterioration is not established unless adverse water conditions are encountered. Thus, at an outage, or during periods of high oxygen content, it is possible to set up a pit in the proximity of a weld or any other disturbed surface.

SUMMARY AND CONCLUSIONS

A severe type of corrosion, local in nature, has occurred in furnace-wall tubes in a number of intermediate and high-pressure boilers. These failures, characterized by the presence of a dark dense layer of magnetic oxide of iron, were embrittled at the inner surface of the tube. More specifically, the failures occurred at the following locations:

1 Welded areas, with and without reinforcements, at both shop and field welds.

2 Away from welded areas, in straight runs of tubing.

3 At cold-worked sections, such as at bends in tubing.

The failures first occurred after operating periods from less than 2 months to 10 years. Generally, the failures occurred after an operating period of about 5 to 6 years. The following conditions have been proposed as causing or contributing to the failures:

FIG. 19 EMBRITTLED AREA OF FIG. 18

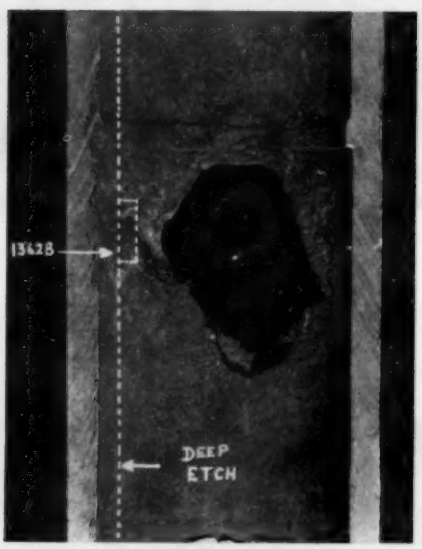

FIG. 20 CORROSION AT SHOP FLASH WELD

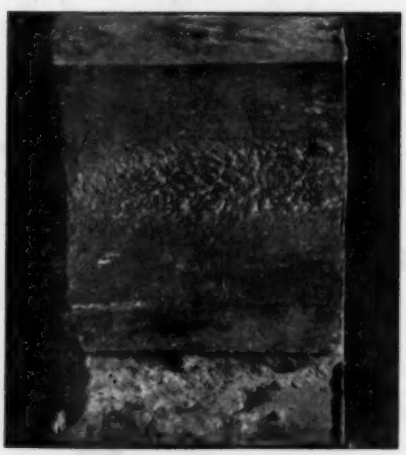

FIG. 21 APPEARANCE OF WIRE-BRUSHED TUBE SURFACE AFTER AN OXYACETYLENE PASS

1 Excessive heat input.
2 Inadequate circulation.
3 Poor deaeration and subsequent addition of oxygen to the boiler at low loads and during outages.
4 Poor chemical control of the boiler and feedwaters and the absence of an oxygen-scavenging agent.

When failures have occurred, a further recurrence of the attack was prevented by one or several of the following items:

1 Improved deaeration.
2 Addition of an oxygen scavenger.
3 Minimizing the introduction of preboiler corrosion products into the boiler.
4 Adjusting boiler-water controls.

In all the cases known to the author, a change in mechanical

FIG. 22 APPEARANCE OF ACID-WASHED TUBE SURFACE AFTER AN OXYACETYLENE PASS

design to alter the furnace-wall heat pattern or fluid flow was not required for prevention of additional failures.

Experience has shown that stressed areas, as at welds or at cold-worked surfaces, are more susceptible to corrosion if adverse operating or water conditions prevail. While there is no evidence to substantiate the theory that backing rings per se will produce failures, welds without reinforcements are less susceptible to attack.

Although the exact role of oxygen is not known in the corrosion mechanism, experience has shown that failures of the nature under discussion occur where poor deaeration is known to take place. Most severe attack has been observed in units where the oxygen in the feedwater periodically is more than 0.05 ppm. The maximum amount of oxygen that can be tolerated cannot be determined from present-day information. It is believed that oxygen may be instrumental in the formation of the initial pit which leads to the accelerated corrosive action.

In view of the field investigations herein set forth, it is recommended that an oxygen scavenger be added to the water supplied to the boiler. The data, in general, indicate that oxygen enters the feedwater systems in quantities greater than is realized by the operating personnel during low-load, night, and week-end operation.

The field studies reported in this paper were made possible by the complete co-operation of a number of operating companies. Their assistance, suggestions, and their willingness to permit test groups to survey plant operating conditions have made it possible to accumulate the necessary data.

Discussion

A. A. BERK.[5] Messrs. Grabowski and Kaufman[6] were not really far apart in their explanation of corrosion in steam-generating tubes. Kaufman shows examples of corrosion produced in the essential absence of dissolved oxygen, and Grabowski reports cases of corrosion resulting from relatively high concentrations of dissolved oxygen that the operators didn't know about because the oxygen was high only at times when analyses were not made. In both presentations corrosion was the fact and the explanation mostly theoretical.

It is axiomatic that corrosion will not occur if the iron-oxide film that separates the steel from the corrosive environment is unpenetrated and unbroken. Erosion attributable to high velocities, impingement, and abrasion can effect destruction of protecting oxide coatings and prevent their ready repair.

In addition, Kaufman suggests that high caustic concentrations produced momentarily by the concentration process incidental to the conversion of boiler water to steam at the steam-generating surface, may be an additional important factor in removing the oxide film.

Grabowski proposes that dissolved oxygen also may damage or prevent the repair of the protecting oxide film. It is of interest that Evans[7] recently suggested a mechanism for such film damage by dissolved oxygen and a second mechanism involving deposited copper. In any case there is little doubt that corrosion would proceed vigorously in the absence of the protecting oxide coating.

It is therefore of considerable importance to understand the mechanism of formation and repair of the oxide coating on steel under boiler operating conditions, and it is expected that such information will be forthcoming from the investigation to be sponsored soon by the Joint Research Committee on Boiler Feedwater Studies.

T. J. FINNEGAN.[8] There has been some tendency to consider this paper on dissolved oxygen and Mr. Kaufman's paper[6] on "bonded oxygen" as controversial. The writer does not believe that any controversy exists. The present author is not questioning the direct attack of the iron by water when the protective film between them has been destroyed. He is simply reminding us that dissolved oxygen, that threat of earlier days, is still a menace, even if a less conspicuous one.

We have found by means of an oxygen recorder that at low loads oxygen to the extent of some hundredths of a ppm may be present in the feedwater in the low-pressure portion of the cycle. We reduce this oxygen to zero within the limitations of our test before we allow it to enter the boiler. Apparently this has been beneficial and the few tube failures which have occurred can be attributed to other causes than dissolved oxygen.

If oxygen can leak into the low-pressure section and if such oxygen is a threat to the boiler, it is obvious that deaeration at the condenser is not sufficient for protection and that a deaerating heater must be installed or resort must be made to chemical deaeration which almost always means sodium sulphite. Chemical deaeration involves little capital cost but it demands frequent, if not continuous chemical feed, and frequent if not continuous

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[5] Chief, Industrial Water Branch, Bureau of Mines, College Park, Md.

[6] This discussion applies also to "Action of Boiler Water on Steel—Attack by Bonded Oxygen," by C. E. Kaufman, W. H. Trautman, and W. R. Schnarrenberger, published in this issue, pp. 423–432.

[7] "Boiler Water Treatment to Prevent Corrosion," by U. R. Evans, *Engineering*, vol. 176, May 8, 1953.

[8] Chemical Engineer, Niagara Mohawk Power Corporation, Buffalo, N. Y. Mem. ASME.

blowdown of the boiler. Mechanical deaeration, on the other hand, enables such simplification of boiler-water control that blowdown is rarely needed and chemical need be fed only once in several weeks.

One of the more disturbing items of the present paper is the suggestion that when the boiler is filled the oxygen in the water can be a potential danger. Deaeration can eliminate that which enters the system during operation but the author is implying that the original filling of the boiler should be with deaerated water. If this should prove necessary it will require considerable ingenuity to do it effectively.

C. E. KAUFMAN.[9] This paper is of interest to us because the author's approach is somewhat different from ours, as expressed in our paper.[6] We have the following specific comments.

In cases of internal corrosion of boiler tubes, we have observed a variety of appearances. Sometimes the affected area is covered with a dense and brittle layer of oxide as the author generally observes, but frequently the surface is free of oxide or is overlaid with a relatively soft and irregular layer of oxide. Frequently, also, hydrogen embrittlement of the metal has not taken place.

Concerning the author's statement of alternate theories of damage, we should like to clarify a point of his which may be intended to represent our thinking. We believe that the presence of more than an optimum amount of free hydroxide in the water may lead to damage under some circumstances, but we know that the lack of hydroxide also can be harmful because a protective oxide film cannot be formed.

The author states that changes in boiler water or better deaeration have resulted in control of damage in nearly every case. But there do seem to be instances where nothing short of physical alterations has achieved relatively trouble-free operation. For example, the introduction of tube spirals in one case and the changing of baffling in another have eliminated difficulties.

With respect to the emphasis to be placed on oxygen as the cause of corrosion, we should like to point out that economizers would be expected to exhibit more damage than the boiler proper since water contacts their surfaces first. Although corrosion is sometimes found in economizers and dissolved oxygen is recognized as the culprit, this damage is very rarely coupled with corrosion in boiler tubes. Furthermore, it would seem that in many boilers, steam-washing would free entering feedwater of the last traces of oxygen.

Sometimes tube failures vary inversely with oxygen content, although we hasten to say that we see no general rule in this. One plant in our experience had numerous tube failures when dissolved oxygen was relatively low; later, with more oxygen, but after acid-cleaning, failures essentially disappeared.

In connection with the author's observations of what appears to be iron oxide in the intergranular cracks at brittle failures, we wonder if oxidation by boiler water could not have occurred after the cracks had been opened up by what we term hydrogen embrittlement.

At one point the author indicates that data pertaining to hydrogen embrittlement have been gathered at room temperature. Our own observations have been made at elevated temperatures.

We believe that the differences of opinion expressed in these papers are a good influence provided they impel us all to a redoubled effort to arrive at the facts.

F. U. NEAT.[10] The matter of dissolved oxygen as a corroding agent in an operating boiler is quite mysterious to me. I cannot understand how there can be any dissolved oxygen there in the first place and, if there were, why it should wait until so far along in the boiler circuit before it attacks and why then it attacks only the bottom (fireside) of a sloping tube or only the fireside of a vertical tube. Furthermore, in its passage through the boiler to its predestined point of attack, why does it not leave a red oxide of iron behind it? And, just to cover all of the other points given as reasons to lay the blame on oxygen, why can we acid-clean a boiler down to bare metal, fill it at least once with fully aerated raw water for a hydrostatic test, leave that water in it for 2 or 3 days, maybe a week or more, then operate a full year with absolutely no change in operating conditions and have not the slightest indication of trouble? Then next year we drain it for inspection, fill it up again one or two or more times with aerated raw water and put it back on the line for another year without the slightest fear of trouble.

Those of us who are in the business of power generation know of the difficulty of getting rid of ammonia. If we have, say, 0.2 ppm or less of ammonia going to a deaerating heater, we will have the same amount of ammonia in the water leaving that same heater. Yet the oxygen, even though it might be as high as 0.2 ppm in this same water to the heater, will be reduced to something around 0.005 ppm in the water leaving the heater. Yet, in the boiler fed with this water, we can assume, and use, the boiler water as a blank for zero ammonia and for dilution water in measuring the ammonia content of the steam. The ammonia must leave before it travels through the boiler circuit as, otherwise, it would not be so completely absent in the boiler water of a boiler having a circulation ratio of 14 to 1 or better. If the boiler will reduce the ammonia so effectively, why will it not reduce the oxygen content even more effectively? Someone might answer that this is due to the difference in pH value of the water. A feedwater pH of 9.0 to 9.3 should be high enough to permit the release of some of the ammonia in a deaerator but the ammonia is not released there. So it is difficult to understand how there can be any oxygen, as such, in the boiler water.

And even though it should be in the boiler circuit as oxygen, it surely cannot be dissolved. If it is not dissolved, then it should attack the top of a sloping tube in preference to the bottom or fireside. Furthermore, why does it attack, preferably, in the hottest zone of the boiler? The author does not say so, but I wonder if in the cases where attack was found in low-heat-transfer areas there was not even more attack in the high-heat-transfer areas or there was more attention paid to maintenance of the high-heat-transfer areas to the neglect of low-heat-transfer areas. For that matter, if it were possible to review those cases in the light of the argument by this writer, an explanation of these apparently "exceptions to prove the rule" might be found for such attack in the odd spots. There still remains the obvious objection to laying the blame on oxygen that this attack is not found in clean boilers and the even more obvious objection found in the absurdity (using the author's term) of blaming oxygen for the attack when there is enough hydrogen being generated to cause hydrogen embrittlement of the steel.

Attributing this sort of attack to dissolved oxygen probably results from a remote similarity in the appearance of this attack to that of known oxygen tubercles in cold-water systems and in older boilers in the days before deaeration. There being this similarity of appearance, thinking has stopped and it is concluded that dissolved oxygen is the cause.

The writer must now revert to his own experience which should point out the difficulty of even assuming that dissolved oxygen is the cause of this corrosion. Oxygen, yes, but dissolved oxygen, no.

Our first B&W type RB boilers were equipped with external-surface-type attemperators for steam-temperature control. The

[9] Director of Research, Hall Laboratories, Inc., Pittsburgh, Pa.

[10] Senior Engineer, Consolidated Gas Electric Light and Power Company of Baltimore, Baltimore, Md.

tubes were vertical U-tubes with the open ends at the bottom and the 180-deg turn on top. Actually, the construction means nothing to this problem so we can neglect it. The important consideration is that these attemperators were excellent settling basins for suspended solids in the boiler water. Approximately 8 months after going into service the first attemperator failed as indicated by boiler water being injected into the steam leaving this attemperator. We took it apart.

Fig. 23 of this discussion shows the top of the tube sheet and the collection of boiler solids. The failure is not shown as it was deep in the bundle and was disclosed and repaired at the factory. The missing thermal-shield nipples, Fig. 24, show the relative location of the failures and, if this is referred to Fig. 23, it will be noted that the failures occurred where the mud was deep.

Fig. 25 shows the iron oxide, magnetic, formed when the failure permitted leakage of boiler water to allow it to contact steel at temperatures higher than saturation temperature so as to cause corrosion of that steel. Naturally, this has but little resemblance to the appearance of the oxide of the failures related by the author, but it shows what an alkaline boiler water concentrating on boiler steel will do to that boiler steel.

Fig. 26 illustrates this quite effectively. And, just to cover another point in the paper, this boiler was carrying the old ASME sulphate ratio but even so did not "by dilution of the chemicals in the film" relieve this attack.

Details of changes in construction to obviate or eliminate this corrosion are unimportant. We attempted better thermal shielding. We "safe-ended" the tubes with Inconel which, heat-treated properly and mechanically not abused, together with the fact of cleaner systems, lasted for as long as 5 years, but even these failed. Some heat-treatments and certain mechanically abused installations of Inconel were good for only weeks instead of months or years.

On the occasion of the first failure of the many we had, an extensive program was instituted to try to find something besides mud as the cause. One metallographic sample was said to have contained copper deep in a crack. There were easier ways of explaining the copper but a theory of low pH was expounded. A chemical sample of the mud might have contained a minute trace of sulphides and decomposition of sulphite was theorized. Oxygen as the cause was not studied as this was practically a base-load boiler with oxygen tests made around the clock at 4-hr periods and the sulphite scavenger under excellent control. Sampling points were established at the attemperator to make the necessary examinations of the water qualities. Nothing of interest was found except that we may have proved a 14 to 1 circulating ratio in the boiler.

When the second and third of the many failures occurred after the regular service periods of approximately 8 months, it was decided to replace the three existing vertical attemperators with new horizontal external attemperators to eliminate the mud difficulty. Boiler installations subsequent to the three external

FIG. 23 COLLECTION OF BOILER SOLIDS ON TOP OF TUBE SHEET

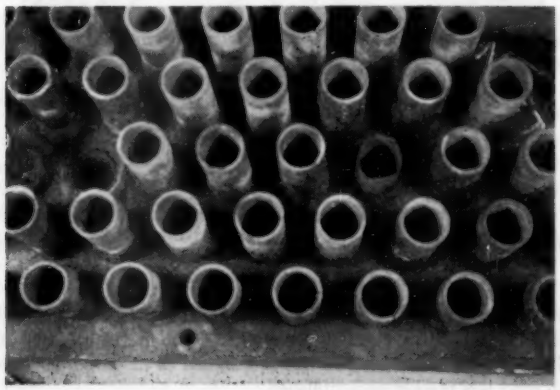


FIG. 24 APPROXIMATE LOCATION OF FAILURES

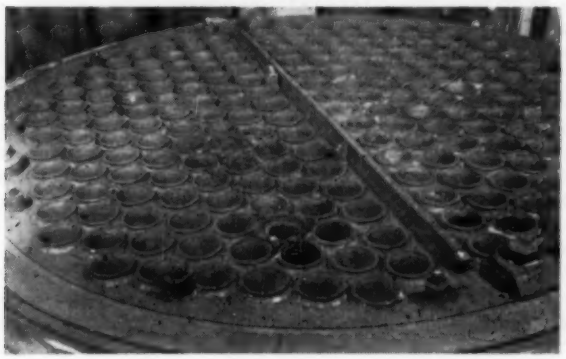


FIG. 25 MAGNETIC IRON OXIDE SHOWN ON TUBE SHEET

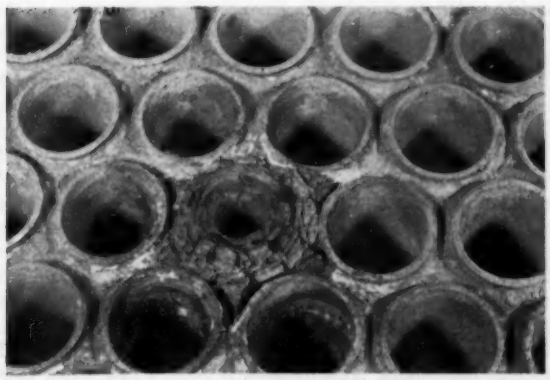


FIG. 26 RESULT OF ALKALINE BOILER WATER CONCENTRATING ON BOILER STEEL

attemperators included horizontal attemperators installed in the lower drum. Then came the third internal installation and a change in design.

This boiler was acid-treated for mill-scale removal. But for some reason or other, despite much care in installation, inspection, and cleanup prior to initial operation, large quantities of mud entered the boiler. The mud was the usual iron oxide and copper. The attemperator failed after about 4 months. The cause of the failure is shown in Fig. 27. The failed tube has been removed but less serious corrosion is shown on an adjacent tube. In operation, the supporting shoe most prominent in this picture is down.

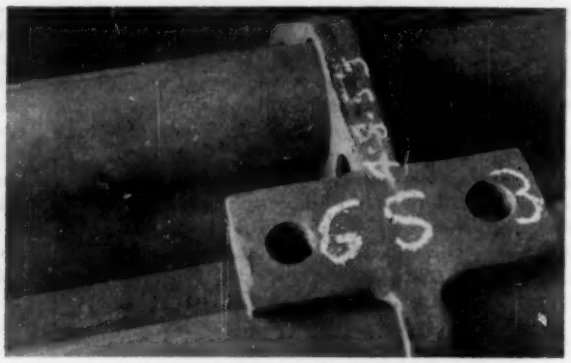


FIG. 27 FAILED TUBE SHOWING SUPPORTING SHOE AS THE CAUSE

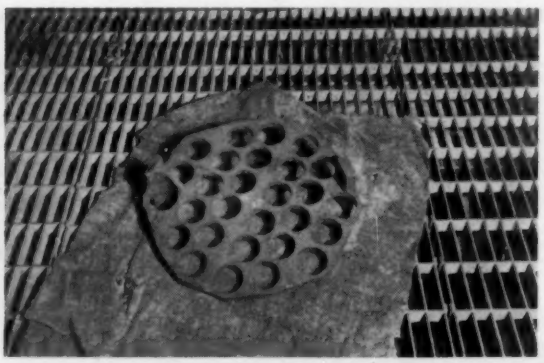


FIG. 28 TOP SIDE OF SUPPORTING SHOE SHOWING CONSTRUCTION

Fig. 28 shows the top side and the construction of this shoe which fitted it close to the failed tube. Mud collected on this shoe and in the annular space between it and the tube, and the tube failed. It is of great interest to this problem that this supporting shoe was not the slightest bit affected by corrosion.

Before discussing the remaining figures of this comment, it is necessary to explain the correlation of this corrosion with that in boilers which the author ascribes to oxygen. And he is not alone on that side of the fence. We shall take only one case as typical.

Steam comes out of the primary superheater at about 830 F. This steam, the amount depending upon the need for attemperation, passes through the tubes. Surrounding the tubes is boiler water which should be at the saturation temperature of around 540 F. The heat-transfer rate is much lower than that found in the furnace parts. Mud gathers and does not allow the boiler water to make continuous washing contact with the tube and the tube-surface temperature must rise. According to the Hall-Partridge film-boiling presentations, it doesn't take much more than 5–7 delta F to raise the concentration of the caustic alkalinity to damaging values and corrosion results. And crevice concentration, as demonstrated by Dr. F. G. Straub in his caustic embrittlement tester of a few years back, is, in all probability, also involved, perhaps due to the delta F function which the Hall-Partridge group expounds.

Now we must look at Fig. 28 again. The support at the left hand is the bottom support. If we turn the picture so this is at the bottom and then imagine the plate in a vertical position, we see the pattern of the tube bundle. It will be noted that there is no vertical free escape path for any steam generated on the lower tubes as a result of the attemperating action. The steam bubbles must rise, crawl around the upper tube thus to blanket the annular space between the hole and the tube. Let's see what happens. Fig. 29 is evidence enough. If not, we can look at Fig. 30. Fig. 31 shows the corrosion of the tube adjacent to the failed tube. This tube was not blanketed as early as the tube that failed.

Fig. 32 shows what we did to these support plates in an attempt

FIG. 29 STAGE IN TUBE CORROSION

FIG. 30 CORROSION IN ADVANCED STAGE

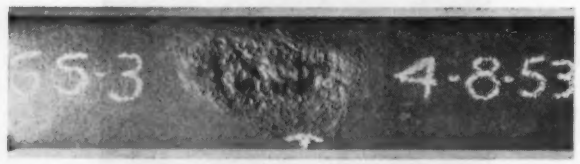


FIG. 31 CORROSION ADJACENT TO FAILED TUBE

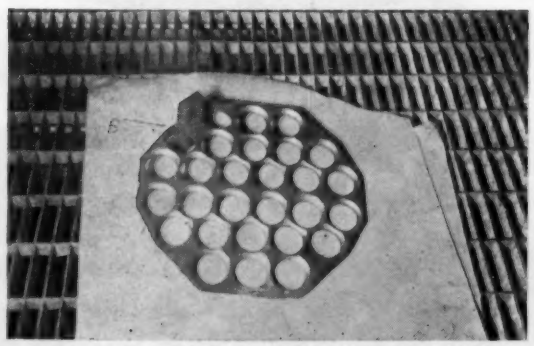


FIG. 32 SUPPORT PLATES CORRECTED TO ALLEVIATE TUBE CORROSION

to alleviate this condition. The holes were tapered toward each side and all burrs carefully removed. The meeting line was slightly rounded. The shoe was redesigned to make piling up of the mud less readily accomplished.

On the basis of what we saw and what we knew about this mat-

ter and about the design of the next older attemperator, we predicted less opportunity for this other attemperator to fail. Contrary, we thought, to our predictions, this older attemperator failed shortly after but at least after a much longer operating period. However, our reputation for prediction is still good in this case. In several spots on this attemperator are small brass tags designating the manufacture and physical characteristics of certain parts. All but one of these tags were securely fastened and snugly fitted to the part. This one tag was fastened at one end only. A little bit of boiler mud collected and we were in trouble. All of these tags were removed just to be doubly sure.

But why did we predict no trouble? Why was it necessary to make such a prediction? We predicted no trouble because the tube pattern is different and there is a vertical escape path for steam generated on the lower tubes. Then the support shoe is designed so as to offer less opportunity for mud to pile up. There seemed to be a much better chance to avoid the conditions which cause corrosion. We had to make this prediction because there were, again, efforts to blame something other than mud as the cause. Acid-cleaning and the presence of minute traces of sulphide in the mud were two. Heat-transfer rate was another. So we selected for our prediction a boiler operating at about the same conditions, with a higher heat-transfer rate in the attemperator and under the same water-treatment schedule. And this boiler also had been acid-treated for mill-scale removal.

Now let's get on with corrosion which is exactly that which the author describes in his paper but which we say has nothing whatsoever to do with dissolved oxygen in the boiler water. Unfortunately, the boilers in which this corrosion occurred operate at 400 psi whereas the author limited his paper to boilers operating at higher pressures. Actually, this makes no difference. This corrosion could, and probably does, occur at lower pressures if conditions are right.

At one of our plants two twin boilers are installed. At the time they were installed they were hailed in the literature as "the thing" in boilers. They have separate deaerators but the water from both deaerators must travel through a common header ahead of the boiler feed pumps. The water must be mixed which must be understood by the reader so that there will be no doubt of the same quality of water going to both boilers. This may be only 98 per cent true, owing to incomplete mixing which might be argued, but it is true enough to eliminate differences in the feedwater.

In the 1930's these boilers corroded and many new tubes were necessary. The writer was not with the company at that time so whether or not this particular occasion was corrosion due to oxygen was not observed personally. The important point of notice is that oxygen was blamed and something was done about it. The boilers were cold, or banked, about as much as they were fired. The deaerators were allowed to cool off vented to the atmosphere. Only caustic soda and sodium sulphate were used in the treatment and total-solids concentration was limited to 600 ppm.

Since oxygen was the suspect, the following changes were made. Sodium sulphite was added to the treatment as intermittent dosage direct to the boiler drum. The lower limit was set at 5 ppm concentration in the boiler water. Coils were placed in the deaerated storage so that any time the units were not running this storage could be kept boiling to give a positive pressure during shutdown to prevent back leakage of oxygen. Where necessary, new tubes were put into the boiler which, in effect, made the highest heat-transfer zone a brand new boiler. The rest of the boiler was mechanically cleaned up. And, since that time, each annual inspection included removal of the handhole plates and mechanical cleaning of all tubes—this latter, because someone also thought that oxygen tubercles as a result of shutdown and draining of the boiler were self-propagating during operation.

The writer, after being employed in 1940 by the company which owns this plant, personally observed that this was carried out. Each year, the tubes were bored until, to the eye, they were as clean as a rifle barrel. The chemical control was checked and rechecked. Oxygen tests were made round the clock at 4-hr intervals. But we were living in a "fool's paradise." We have since found out that mechanical turbining of a boiler will not remove the multilayered magnetic oxide-copper scale (baked sludge?) such as is formed in low make-up, low-alkalinity (65 ppm as sodium carbonate minimum) boilers. Such scale is only polished by mechanical cleaning.

Then we were jarred out of our complacency. In the 1949 annual inspection of the second of these boilers, extensive corrosion was found. In a few more days, weeks, or months, perforations would have occurred and observations would have been made exactly as the author describes in his paper. Lacking perforations, there were only the deep pits filled with dense plugs of magnetic iron oxide layered with copper and mounded. A red cap which one could assume to be red oxide of iron but which was actually a mixture of copper and magnetic oxide completed the picture to indicate that dissolved oxygen was the culprit. Those who had seen the corrosion in the 1930's and to whom this story was told or who saw for themselves said it was the same as the earlier variety. Lacking the convenience of hydrogen recorders, we have no idea as to when the corrosion started. We do know that it was not there on the occasion of the previous inspection and that the tubes were bored on the occasion of that previous inspection.

Let's look at the oxygen story. These boilers went into service in December, 1926. Seven years later, in May, 1933, both boilers were pitted which resulted in essentially starting again with a new boiler with full knowledge of the terrors of oxygen. Sixteen years later, in May, 1949, only one boiler was pitted. Bear in mind that the same water must go to both boilers even though there are two deaerators. There are differences in service hours, as there are two turbines as well as two boilers, and there are the other unknown factors which prevent two identical boilers from behaving like true twins. After the first failures everything then known was done to stop oxygen and we started again with new boilers.

On the occasion of this second failure we again heard all about oxygen. But this time we also heard we were adding sulphite improperly and that it should be added to the feedwater continuously instead of periodically to the boiler. What we did about this is a little ahead of the story but must be told now. We instituted a program of chemical testing for oxygen, using the ASTM Referee method with dead-stop end point. Those who were broken in on the operation were not permitted to do the job until duplicate samples taken simultaneously were in consistent agreement to one split drop of N/1000 titrating solutions. That agreement was found to be possible, or rather was made possible, by refining the technique of the method and not the method itself. It was necessary to use special grease on stopcocks, to clean the sampling flasks with solvents prior to each sampling, to destroy flasks which gave consistently wrong answers, to pay strict attention to elapsed time of transfer of sample to titrating vessel plus time of titration and to use techniques permitting least opportunity for dissolving new oxygen in the sample during and after release from the flask. Then, if no sampling flasks were broken, duplicate samples taken simultaneously agreed within one split drop of N/1000 titrating solutions. You can calculate that out to show that it is much better precision than ASTM will grant is possible.

So much for the method. Samples were taken every 2 hour. And we found, chemically, what was soon proved with a Cambridge

oxygen recorder. As is usual in any power plant, dissolved oxygen was higher on start-up periods. We also found that dissolved-oxygen guarantees should be based on the amount of oxygen going to the heater. But the oxygen values weren't too bad. What is more important is that there is no reason to believe that the performance was any different from the day they first started operation or, rather, since the time those changes were made which usually are made to a deaerator in an effort to meet the guarantee.

Furthermore, we must keep in mind that only one boiler was corroded. So therein lies what can be a very strong indication that oxygen is not the story. When the corrosion was found, we telephoned Mr. Arthur Guy of Commonwealth Services, who was then located in Ohio. We asked him if he would please get on a plane with his scale detector and point the plane toward Baltimore, which he did. After instructing us, he was kind enough to loan the instrument to us to complete the survey. We found 0.010 to 0.012 in. of scale (baked sludge?) as a general thing with localized areas up to 0.030 inches. This was No. 1 boiler, the pitted boiler.

Just a few weeks prior to our looking at No. 1 boiler, No. 2 boiler was out for its inspection period and for the first time since the tube failures in 1933, it was not turbined. We had noted, however, that we should consider an acid-cleaning in 1950. So, after our findings in No. 1 boiler, and badly pitted tubes had been replaced, others had been repaired by welding and grinding and still other pits were hopefully left to their own devices and the boiler was acid-cleaned, we reopened the No. 2 boiler. The scale detector showed only 0.008 to 0.009 in. as a general thing, with local indications up to 0.020 in. Note carefully the difference in scale-thickness indications. This boiler was also acid-cleaned.

Mr. Arthur Guy told us that, in his experience, when the scale detector indicates 0.010 in., a boiler should be acid-cleaned and that pits are always found where scale readings are around 0.015 in. in the high-heat-transfer zones. Dowell people said that no boiler has shown corrosion when the acid-soluble material calculated back to the boiler heating surface is less than, as I remember it, 0.006 in. All corroded boilers calculate back to thicker scale.

Both boilers were returned to service with no changes whatsoever in operation or treatment. On the basis of the findings with the dissolved-oxygen recorder we tried to improve the venting and drainage of the deaerators but little was accomplished in so far as oxygen was concerned. Since that time both boilers were treated for copper removal and No. 1 boiler was acid-cleaned again in 1953 because of spotty readings with a scale detector. No. 2 boiler showed only 0.002 in. with the scale detector in 1953, so it was not cleaned. There is no new corrosion, the old pits are still there, inactive, and the welded spots are as they were left at the end of the 1949 repair. And to cover the "welding damage" mentioned by the author, these were electric welds with no heat-treatment, some as close as 2 in. or less to the rolled joint.

But in so far as the writer is concerned, dissolved oxygen, as such, is not to be worried about in an operating boiler. Keep your boiler clean and you won't have pitting unless, of course, you allow it to cool off full of water and vented to the atmosphere. But, even then, if the subsequent tubercle scale is not thick enough, it will not propagate itself when the boiler is returned to service. And if you turn to your physical chemistry of oxygen-cell corrosion, you will know why this is so.

In most states, including ours, the law states that you must open up your boilers annually for inspection. That is the worst thing which can be done to a boiler, other than to let it get dirty, but it must be done. Each time it is done, tubercles are started, many where they can't be seen. And, in the modern boiler, you cannot send a tube cleaner through to knock off these tubercles. And, after the internal inspection, a hydrostatic test is applied. If evaporated water is in short supply, and it usually is, the boiler is filled with air-saturated raw water with a little condensate floated over the top for a little better care of the superheater. Maybe that water will stay there for a week but at least for a day. And this water is warmed to about 70 F just to make sure that tubercles are started. But, because of the preponderance of boilers not experiencing this corrosion trouble, it is quite evidence enough that we don't need to look there for the trouble.

And just at this time I'd like to take particular exception to the author's statement, "In nearly every instance, chemical changes in the boiler water and improvements in the feedwater system, such as deaeration, have resulted in control of even the most severe corrosive activity." Let's get a few more years' experience on his case histories. Let's go even a little further. Let's keep watch on the scale thickness and everyone knows if we put down a layer of scale (or paint) we can stop oxygen attack. Then let's do nothing about reduction of this scale thickness.

So what causes this trouble? First, there must be scale or baked sludge thick enough, the thickness depending upon the rate of heat input and operating pressure. We all know that if this scale gets thick enough, the tube will bulge and blow due to overheating. But what happens before that time? The scale, or baked sludge, being thick enough, the tube-wall temperature must be elevated. I did not say that the tube would be overheated to the point of metallographic indications of higher temperatures. Only 5 to 10 deg over saturation temperature is enough. Then, because of occlusions in the scale of fibers, wire, or other usual debris or to the difference in thermal-expansion rates of the scale and the boiler metal, a crevice forms to allow boiler water to contact this higher-temperature surface but not to a continuous washing contact and we are in trouble. Bear in mind that initial contact doesn't have to be on the metal. Somewhere inside the oxide film is a point of temperature high enough above saturation temperature to cause the boiler-water alkalinity to concentrate enough to destroy this oxide thus to soon contact the metal, or the initial contact can be clear down to the metal.

It may be argued that this corrosion occurs in boilers where no chemicals are added, as the author states. But he also mentions a pH of 8.5. I have no doubt that it was higher or lower at times but any solution containing sodium and carbonate ions (both are nearly ubiquitous) will do and at a pH of 8.5 these ions were there. These ions being there, no matter how low the concentration, a delta F of only 5 to 10 deg will concentrate them to exactly the same concentration as it would a boiler water initially at any other higher pH. The only difference would be the time factor. And the sodium carbonate would hydrolyze to sodium hydroxide if it was not sodium hydroxide in the first place. This is only a matter of physical chemistry laws and is not simply because Hall and Partridge formulated the delta F function. They only gave it a short name and demonstrated that which was known to be so.

The author mentions corrosion in boilers on the "phosphate-equilibrium" treatment with which we have no quarrel. It only proves that this treatment is not the final answer to our boiler troubles. The phosphate equilibrium treatment is a device to obviate caustic embrittlement, a most peculiar form of boiler trouble. However, that doesn't remove the possibility of this sort of corrosion.

Our boilers, and most all boilers, exhibit phosphate hide-out troubles after acid-cleaning. In our case, we can get out of this trouble by removing all phosphate from the treatment and from the boiler, then operating the boiler at a higher than usual caustic alkalinity under full boiler load for 1 or 2 days. Phosphate can then be returned to the treatment and the alkalinity reduced to normal values and no further trouble with hide-out is found.

Just a word of warning on another subject is found in that silica goes with the phosphate. If you have phosphate hide-out you also have silica hide-out. On only one boiler, and this is being

done now so the result cannot be reported, have we had to go back for a second try. Possibly we were not entirely free of silica and phosphate and possibly the load wasn't high enough long enough on the first try. It's the first of eight boilers which missed in the cure.

This tends to prove that there may be a reactive film in the boiler. Others have cured this hide-out simply by pumping phosphate until the reactive film is satisfied. If it is not a reactive-film phenomenon, then those who had the nerve to pump all this phosphate would certainly have lost their boilers. But if this reactive film exists at zero delta F, then who knows or who has attempted to find out what happens in the phosphate-equilibrium treatment when a delta F exists? The author exposes a case history which must be looked at from other than the dissolved-oxygen angle.

The author knows of at least one instance where this corrosion occurred and where the dissolved oxygen was as low as can be found in a power system. Furthermore, it was in a no-treatment plant but where the boiler water pH was around 10.0 or 10.5. At that plant the entire effort was directed at there being excellent deaeration and valves were even sealed with deaerated water. There appears little reason to believe that the quoted oxygen figures of the operators of this plant are not reliable. But the scale in the boiler was too thick and the boiler corroded.

So it is not dissolved oxygen in the boiler which does the pitting. We have by exposure of our experience, described how each of the failures the author mentions was caused by collection, deposition, and hardening of scale or boiler sludge. Why they occurred in such odd and various locations is explained fully. If this comment serves only to enable more people to realize that Hall and Partridge did not establish a law and that they only gave a short name to a well-known physical-chemistry law so that these same people will look for the true cause of their trouble, it has done its work.

Look for your dissolved-oxygen trouble in your preboiler system. The role it plays, in company with its companion in crime, carbon dioxide, to permit iron and copper oxides or salts to enter the boiler circuit there to deposit, to permit a delta F to develop is the role of oxygen. If corrosion is stopped in the preboiler system or if the boiler is kept clean or cleaned up when necessary, this sort of pitting will not occur. Don't trust your eye or a tube-boring machine to check for scale. Use a scale detector.

E. R. Woodward.[11] The author focuses attention on the actual amounts of dissolved oxygen found to be present in boiler water under various circumstances, and these data are set forth very clearly in his curves. The most startling increases in oxygen content are shown in Fig. 3.

In each case the oxygen content reaches or exceeds 0.5 ppm. This is 100 times as great as the 0.005 ppm oxygen content of deaerated boiler feedwater, so often quoted by operating power engineers as the oxygen content of their boiler water.

Regarding the recommendation that an oxygen scavenger be added to the water supplied to boilers, the writer wishes to point out that hydrazine hydrate and various hydrazine salts are now being manufactured commercially in the United States, and are already in use for this purpose. Indications are that these compounds are valuable as oxygen scavengers in operating boilers, and also in the protection of idle boilers and other equipment which is laid up completely filled with water.

Author's Closure

The corrosion problems that have been illustrated in this paper were found in vertical sections of the furnace-wall tubes. The physical appearance, the incidence of failure, and the metallurgical structure of the corroded areas have been defined clearly. Oxygen has been indicated as being a factor in the corrosion process in the cases cited by the author. Amelioration of attack had been achieved without altering mechanical design or changing the heat-transfer pattern in these cases.

At the same time, the author did not wish to imply that all corrosion problems in a boiler are attributed to the presence of oxygen. It should be quite clear that metal attack in nearly horizontal slag-screen tubes, or in an attemperator which is associated with the action of concentrated boiler water on the metal is not related to the specific and local corrosive action described.

Since the basic reaction of iron and water, over the range of pressures below the critical point and those above the critical are not clearly understood, it would appear foolhardy to indicate that the prime factors controlling the corrosive action in the boiler are similar in every case.

Mr. Berk, in his comments, has defined the problem which confronts the Joint Research Committee on Boiler Feedwater Studies. We must determine the mechanism of repair and formation of the oxide coating on steel under boiler operating conditions. As shown in the paper under discussion, and suggested by Evans, oxygen may damage or prevent a repair of the protecting oxide film. Mr. Berk's comments are indeed appreciated. His views offer a constructive appreciation of the factors which must be investigated.

The author has illustrated periods of operation when oxygen can enter the feedwater system from the low-pressure portion of the cycle. Mr. Finnegan reports a similar instance when oxygen could enter the feedwater cycle if it were not detected by a continuously recording analyzer. Mr. Finnegan's point that the use of a deaerating heater is a positive means of reducing the oxygen problem is a good one. In testing 17 utility stations, the author found that only 2 systems out of 10, which maintained all deaeration in the condenser, were consistently low in oxygen at all conditions of operation.

As indicated in the paper, it is possible that an initial pit in the tube metal can form by the action of oxygen during an outage or during operation. The latter pit can become the focus of rapid attack at full-load operation. The author can appreciate Mr. Finnegan's concern at his implication of the advisability of starting up a boiler with a low-oxygen water. The danger of oxygen during an outage or start-up period can be minimized by the addition of an adequate reserve of a reducing agent as sulphite or hydrazine. Private reports from Europe, in the author's possession, indicate that hydrazine has afforded the boiler metal an excellent degree of protection when it has been added to the water during an outage period.

Mr. Kaufman presents a number of interesting and thought-provoking statements. The author concurs with Mr. Kaufman's statement that a proper addition of caustic assists in the preservation or perhaps the establishment of a protective oxide film. However, numerous boilers are operating successfully without the addition of any chemicals in the feedwater or boiler-water circuits. Evaporated or demineralized make-up is supplied to tight systems where the only solids in the water consist of the reaction products of water with the metal surfaces. The incidence of failures in these cases is below those observed in treated systems. Deaeration of the feedwater to these units is excellent at all times. Thus it is apparent that although the caustic alkalinity acts to insure a formation of protective film, it may not be the prime factor in the control of the attack of pure water on steel in a steam generator.

As mentioned previously, the author does not suggest that oxygen controls the corrosive activity of water in a tube that is slightly inclined from the horizontal plane. As Mr. Kaufman

[11] Mathieson Chemical Corporation, Baltimore, Md.

indicates, spirals often are introduced into these tubes in an attempt to force the water into contact with the heated portion of the metal. The metal is overheated because the turbulence of the water supply is insufficient to insure a wetting of the heated surface. To compare this action with the local attack discussed in this paper would be wrong in principle and fact. It must be remembered that design alterations were not utilized in correcting the corrosive action in the cases cited.

The author believes that one of the basic obstacles to the understanding of a fundamental problem is that of generalization and the formation of conclusions from what appears to be logical reasoning that is not entirely founded on facts. Until the intensive survey of feedwater systems was undertaken, the author was led to believe that the oxygen content in the feedwater supplied to the boiler is always less than 0.01 ppm and more often 0.005 ppm. These concepts were dispelled by tests. Rather than applying reason and logic to the solution of problems, every effort should be taken to determine the facts.

Mr. Neat offers his experiences, comments, deductions, and conclusions on the subject of corrosion. The author believes that many of Mr. Neat's comments relative to salient points in the paper have not been evaluated properly and completely. One such point concerns the comments relative to the pH effect in the nontreatment plants cited by the author.

Conclusions based on limited facts, when expressed too generally, can produce uncertainty and doubtful practices in the field. That engineers are operating boilers without any chemical treatment of the water is proof of the indecision among personnel as to the correct procedure to follow. The latter operators seem to have encountered far less trouble than reported by Mr. Neat as existing in his system.

The author wishes to express his appreciation to the discussers for their timely and pointed comments relative to the subject under discussion. This general interest when applied to the media of research should result in the elimination of serious tube problems which lead to a loss of operating potential in a power plant.

Experimental Boiler Studies of the Breakdown of Amines

By C. JACKLIN,[1] CHICAGO, ILL.

A report of the results obtained when several types of amines were tested for breakdown to ammonia in an experimental boiler. Test pressures ranged from 250 to 2500 psi and temperatures from 400 to 1200 F. Test results indicate that morpholine, which has been used successfully at 1600 psi and 1000 F, also shows relatively good stability at 2500 psi and 1200 F.

Introduction

IMPROVED methods of feedwater preparation for high-pressure steam-generating plants have reduced the amount of deposit-forming materials to low values. However, the circulation of pure condensate and/or make-up water through the storage tanks, feedwater piping, pumps, feedwater heaters, turbines, and condensers results in appreciable pickup of iron from this predominantly ferrous system. This iron tends to accumulate on the heat-transfer surfaces of the boiler where it may cause overheating and tube-metal deterioration.

Carefully controlled experiments have shown that the pickup of iron by pure water is at a minimum when the pH of the water is approximately 9.0 to 9.2.[2] The use of nonvolatile inorganic chemicals in the feedwater and condensate to increase the pH, and thereby reduce the pickup of iron, is not practical in many cases since these chemicals concentrate in the boiler water and must be removed through the blowdown system.

The ideal solution to the problem would be a volatile, alkaline, organic chemical which could be added to the feedwater or condensate in reasonable amounts to increase the pH and thereby decrease to a minimum the amount of iron which would be picked up. It should volatilize readily from the boiler water and condense in the initially formed condensate to give protection to all parts of the system. It also should be stable at the highest temperatures encountered in the superheater.

At the time this study was started, cyclohexylamine and morpholine had been used for controlling the pH and iron pickup in low-pressure systems with good success. However, their reported use was limited to systems operating below 400 F.[3] Success with the use of amines for condensate-line-corrosion control in low-pressure systems and the increasing need for the control of iron pickup in high-pressure systems focused attention on the need for amines or similar materials which would be satisfactory for use in boilers and turbines operating at high pressures and superheat temperatures. To qualify for this service a material must be relatively stable at temperatures up to 1000 F and must raise the pH to a protective level when used at a reasonable dosage.

A research program was planned to test a number of amines and related materials for possible use as corrosion-control agents in high-pressure boiler systems. Since the stability and pH increase were considered most important, the test equipment and procedure were set up to measure these two properties.

Experimental Equipment

A laboratory-size, experimental high-pressure boiler which has been used for many studies[4] of deposits in boilers was used for the first series of tests which were run at saturation temperature. A superheater was added to this boiler to obtain the higher steam temperatures used in later tests. This superheater consists of a 10-ft length of $^1/_4$-in-OD × $^1/_{16}$-in-wall stainless-steel tubing closely wound on a $1^1/_4$-in-mandrel. Heat is supplied to the superheater by replacing the mandrel with a wire-wound resistance-type heating element of about 3 kw capacity. Steam from the boiler enters the lower turn of the superheater tubing, passes upward through the tubing which encircles the heating element, over a thermocouple well, and then into a second stainless-steel coil immersed in cooling water. Pressure in the boiler and superheater is controlled manually by a needle valve in the condensed-steam line. Temperature of the superheated steam is measured at the outlet of the superheater with a chromel-alumel thermocouple and temperature recorder. Temperature control is by manual adjustment of a variable-voltage transformer which supplies power to the superheater heating element. The entire superheater and connecting lines are heavily insulated to reduce heat losses and maintain uniform temperature conditions.

The condensed steam flows from the pressure-control valve through stainless-steel tubing and a short length of rubber tubing to a flow cell which contains the electrodes for measuring pH. Samples collected from the overflow of the pH cell are tested for ammonia by direct Nesslerization.

Boiler-Water Conditions

Feedwater was prepared by passing Chicago tap water through a two-bed deionizing unit and then through a small mixed-bed deionizing column to obtain high-purity water. This feedwater was treated with sodium hydroxide, sodium chloride, disodium phosphate, and sodium sulphite to give the following boiler-water readings when operating at a concentration ratio of 10:

Boiler-water composition	Ppm
Methyl-orange alkalinity (as $CaCO_3$)	100
Chloride (as NaCl)	50
Phosphate (as PO_4)	10
Sulphite (as Na_2SO_3)	10

The material to be tested was added to the feedwater at the rate of 34 ppm; the test was started and allowed to run for several

[1] Director of Engineering Research, National Aluminate Corporation.

[2] "Prevention of Corrosion and Metal Attack in the Steam Water Cycle of the Steam Power Plant," by F. G. Straub and H. D. Ongman, *Corrosion*, vol. 7, 1951, pp. 312–315.

[3] "The Causes and Prevention of Condensate Return Line Corrosion," by R. T. Hanlon, Proceedings of the Midwest Power Conference, Chicago, Ill., April, 1948.

Contributed by the Joint Research Committee on Boiler Feedwater Studies and the Power Division and presented at the Annual Meeting, New York, N. Y., November 29–December 4, 1953, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, November 27, 1953. Paper No. 53—A-240.

[4] "Experimental Studies of Boiler Scale at 800 Psi," by J. A. Holmes and C. Jacklin, Proceedings of Fourth Annual Water Conference Pittsburgh, Pa., November 1–2, 1943.

hours until equilibrium was reached before any readings were taken. To avoid any build-up of ammonia in the system all tests were run with 100 per cent make-up of feedwater. All of the condensed steam was discarded. A separate test was run for each material tested. However, results at different pressures and temperatures on the same material were obtained by operating at one condition until equilibrium was reached, collecting the data, and then changing to the next pressure or temperature without restarting the test.

The materials tested were suggested by members of the research organization and other interested personnel. The primary objective of the first series of tests was to make a preliminary survey of classes of materials which might be suitable for control of iron pickup.

Test Results

Table 1 shows the pH and ammonia content of saturated-steam samples when different nitrogen-bearing organic compounds were added to the boiler feedwater at the rate of 34 ppm. It is evident from these results that the different compounds vary rather widely in their behavior under these conditions. The pH readings range from 6.60 to 10.00 and the ammonia concentrations range from 0 to 19.5 ppm.

On the basis of the results from these preliminary screening tests, 2-diethylaminoethanol was selected for additional testing at higher steam temperatures.

Table 2 shows the pH and ammonia concentrations obtained on the condensed-steam sample while the test boiler was operating at 1500 psi and the superheated-steam temperature shown. 2-Diethylaminoethanol was maintained at 34 ppm in the boiler feedwater. These results are shown in graphical form in Fig. 1.

The rate of breakdown of 2-diethylaminoethanol increases gradually as the superheat temperature increases up to 800 F. Between 800 and 900 F the rate of breakdown increases sharply and the amount of ammonia formed becomes excessive. For practical purposes the upper steam-temperature limit for this material is approximately 800 F. The urgent need for a material suitable for use in systems operating in the 800 to 1000 F range prompted investigation of the stability of morpholine even though its reported use was limited to 400 F or lower.

Table 3 shows the pH and ammonia content of the condensed-steam sample while the test boiler was operating at 1500 psi and the superheated-steam temperature shown, while morpholine was maintained at a concentration of 34 ppm in the boiler feedwater. The maximum superheat temperature reached in this test was 985 F. Table 4 shows the results of a recent test with morpholine in which the superheated-steam temperature was run up to 1200 F. The results given in Tables 3 and 4 are shown in graphic form in Fig. 2.

The results shown in Table 5 were obtained under the same conditions as those in Table 4, except that the boiler pressure was

TABLE 1 STABILITY OF AMINES AT SATURATED-STEAM TEMPERATURES

Dosage—34 ppm amine in feedwater

Pressure	500 psi		800 psi		1500 psi		2500 psi	
Saturation temperature	467 F		518 F		596 F		669 F	
	pH	NH_3, ppm	pH	NH_3, ppm	pH	NH_3, ppm	pH	NH_3, ppm
Blank	..	..	..	..	5.42	0.16	..	..
					5.40	0.14		
					5.30	0.12		
Ethylenediamine	..	..	..	..	9.20	1.46	..	..
					8.90	3.89		
					9.15	2.92		
Propylenediamine	..	..	..	..	9.45	6.8	..	..
					9.55	7.8		
					9.55	3.9		
2-Dimethylaminoethanol	..	..	..	..	9.40	1.46	9.30	0.15
					8.85	1.09	9.15	0.19
							9.00	0.19
2-Diethylaminoethanol	..	..	9.05	0	8.60	0.02	9.5	0
			9.10	Tr	7.45	0.02	9.65	0
			9.25	0.01	9.15	0.01	9.00	0
2(2-Dimethylaminoethoxy) ethanol	..	..	..	..	9.02	0.49	..	..
					7.70	0.73		
					8.25	0.97		
2[2(2-Dimethylaminoethoxy) ethoxy] ethanol	..	..	..	..	8.45	0.57	..	..
					6.95	0.81		
					7.00	0.68		
1-Dimethylamino-2-propanol	..	..	..	..	9.50	0.65	..	..
					9.30	0.65		
					9.10	0.73		
Monocaproamide of ethylenediamine	..	..	..	..	6.80	0.65	..	..
					6.70	0.81		
					6.60	1.13		
Cyclohexylimido di-2-propanol	..	..	..	..	7.25	0.49	..	..
					6.65	0.81		
					6.70	0.79		
3-Dimethylaminopropylamine	..	..	..	..	9.35	0.97	..	..
					8.90	0.73		
					9.15	0.73		
3-Isopropoxypropylamine	..	..	..	..	9.45	0.61	..	..
					9.20	0.73		
					9.10	0.85		
Guanidine sulphate	..	..	..	..	8.95	16.2	..	..
					8.95	13.6		
					9.00	19.5		
Morpholine	8.3	0.34	8.5	0.27	8.15	0.49	8.20	0.75
	8.2	0.34	8.5	0.24	8.05	0.44	8.25	0.72
	8.4	0.34	8.5	0.22	7.90	0.49	8.37	0.73
Cyclohexylamine	..	..	..	..	9.60	0.29	..	..
					9.50	0.29		
					9.30	0.34		
Piperazine	..	..	..	..	10.00	2.00	..	..
					10.02	2.10		
					10.00	2.10		

TABLE 2 STABILITY OF 2-DIETHYLAMINOETHANOL IN SUPERHEATED STEAM

(34 ppm in feedwater; pressure, 1500 psi)

Steam temp, deg F	Condensed steam pH	Condensed steam NH_3, ppm
596	9.3	0.12
596	9.2	0.19
596	9.2	0.19
596	9.3	0.19
596	8.6	0.12
596	8.0	0.12
596	9.4	0.12
596	9.2	0.24
620	9.2	0.15
640	9.1	0.19
695	8.9	0.15
700	8.7	0.29
700	7.5	0.49
710	9.2	0.34
710	8.9	0.49
705	8.6	0.49
785	8.4	0.68
785	8.6	0.58
750	7.8	0.68
850	7.3	0.73
870	8.9	0.97
770	9.0	0.58
770	8.2	0.58
805	7.9	0.73
840	9.0	0.73
880	8.3	0.10
870	...	0.97
890	...	1.70
890	...	1.10
890	7.5	1.46

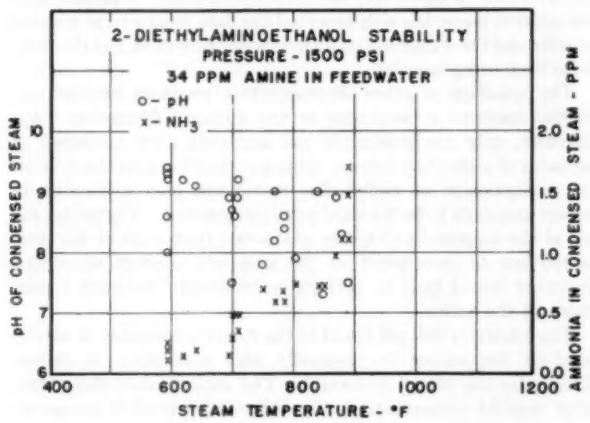


FIG. 1

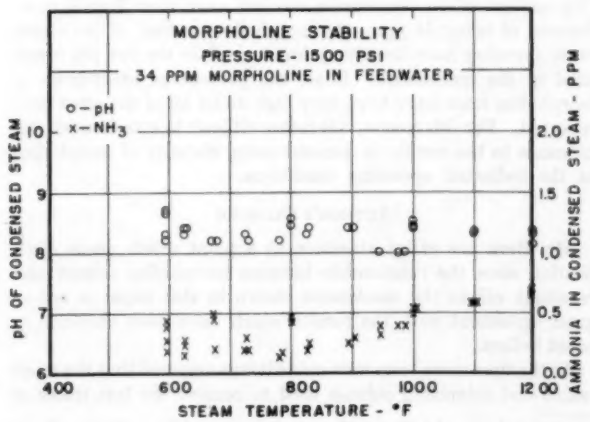


FIG. 2

TABLE 3 STABILITY OF MORPHOLINE IN SUPERHEATED STEAM

(34 ppm in feedwater; pressure, 1500 psi)

Steam temp, deg F	Condensed steam pH	Condensed steam NH_3, ppm
596	8.2	0.27
596	8.2	0.09
596	8.3	0.19
596	8.3	0.10
623	8.3	0.24
624	8.4	0.15
622	8.4	0.27
628	8.4	0.29
675	8.2	0.19
665	8.2	0.08
670	8.2	0.49
675	8.2	0.44
730	8.2	0.19
725	8.3	0.19
725	8.3	0.29
780	8.2	0.15
785	8.2	0.17
785	8.3	0.15
780	8.3	0.19
826	8.3	0.19
829	8.4	0.19
829	8.4	0.25
825	8.4	0.25
894	8.4	0.25
902	8.4	0.29
945	8.0	0.38
945	8.0	0.31
975	8.0	0.38
985	8.0	0.38

TABLE 4 STABILITY OF MORPHOLINE IN SUPERHEATED STEAM

(34 ppm in feedwater; pressure, 1500 psi)

Steam temp, deg F	Condensed steam pH	Condensed steam NH_3, ppm
596	8.65	0.40
596	8.66	0.39
596	8.65	0.42
800	8.50	0.45
800	8.50	0.45
800	8.50	0.44
1000	8.52	0.52
1000	8.43	0.50
1000	8.40	0.52
1100	8.29	0.60
1100	8.30	0.59
1100	8.32	0.59
1200	8.36	0.66
1200	8.15	0.68
1200	8.30	0.66

TABLE 5 STABILITY OF MORPHOLINE IN SUPERHEATED STEAM

(34 ppm in feedwater; pressure, 2500 psi)

Steam temp, deg F	Condensed steam pH	Condensed steam NH_3, ppm
669	8.20	0.75
669	8.25	0.72
669	8.37	0.73
669	8.38	0.72
1000	8.46	0.75
1000	8.48	0.73
1000	8.50	0.74
1100	8.55	0.77
1100	8.52	0.78
1100	8.52	0.78
1200	8.58	0.81
1200	8.58	0.81
1200	8.56	0.81

2500 psi instead of 1500 psi. These results are shown in graphic form in Fig. 3.

During the early part of this investigation tests with 2-diethylaminoethanol were made at 400 and 900 psi and steam temperatures up to 800 F. Morpholine also was tested at 250, 500, and 800 psi, and steam temperatures up to 800 F. The results of these tests have been omitted since they showed no significant deviation from the results at 1500 psi.

DISCUSSION OF RESULTS

The urgent need for finding a material to use in plants which were in trouble with iron-oxide deposits did not permit a thorough and detailed study of each of the materials tested. The number of amines which could be tested also was limited. As a result, the

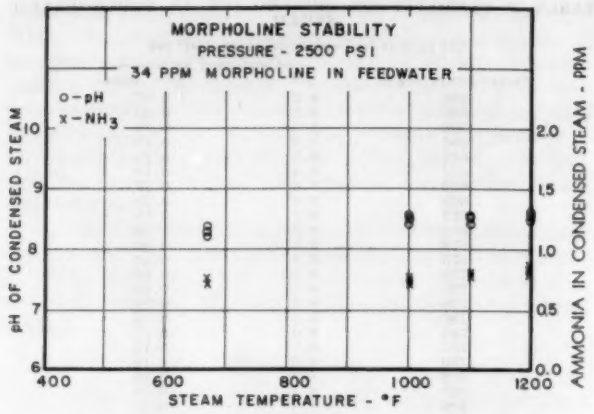


FIG. 3

data collected lack some of the precision and completeness usually associated with a well-developed research project.

Under the conditions present in the first series of tests, cyclohexylamine, 2-diethylaminoethanol, and morpholine gave less than 0.5 ppm of ammonia in the steam. Cyclohexylamine was eliminated from further study because its distribution ratio is such that it has a low solubility in the initially formed condensate in a system. In contrast, the distribution ratio of morpholine is such that it concentrates in the initially formed condensate.

Since rather vague but persistent reports intimated that morpholine was unstable at high temperatures, the test work was first concentrated on 2-diethylaminoethanol.

The tests with 2-diethylaminoethanol at steam temperatures up to 890 F showed that this amine is relatively stable up to 800 F. Above 800 F the amount of breakdown to ammonia increases rapidly as the temperature increases.

The laboratory results on this amine were confirmed by a number of field applications below 800 F where it was used for increasing the pH of the condensate system. The amount of iron pickup was reduced materially and magnetic iron-oxide deposits in the boiler were practically eliminated.

Meanwhile, experimental boiler tests on morpholine showed only a slight increase in the amount of breakdown to ammonia at steam temperatures up to 985 F.

Since morpholine was both more stable at high steam temperatures and lower in cost, it immediately replaced 2-diethylaminoethanol for condensate pH control in high-pressure high-temperature systems. Several years of experience in field applications have verified the laboratory results.

Recent interest in volatile materials for pH control at much higher temperatures and pressures prompted us to reactivate our test program and check the stability of morpholine at pressures up to 2500 psi and steam temperatures up to 1200 F, the upper operating limits of our present equipment.

The absence of any sharp increases in the amount of breakdown of morpholine to ammonia under these conditions indicates that it is still relatively stable.

The effect of pressure on the breakdown of morpholine is shown by comparing the results in Figs. 2 and 3. Increasing the boiler pressure from 1500 to 2500 psi causes an increase of about 40 per cent in the amount of ammonia formed. Pressure had very little effect in the range between 250 and 1500 psi.

SUMMARY

Different amines showed widely different amounts of breakdown to ammonia when tested in an experimental boiler at 1500 psi and saturation temperature. Of the fifteen materials tested, cyclohexylamine, 2-diethylaminoethanol, and morpholine showed relatively low breakdown rates. Cyclohexylamine was not tested at higher temperatures because its usefulness is limited by its low solubility in the initially formed condensate in a system.

2-Diethylaminoethanol showed good stability at 1500 psi and steam temperatures up to 800 F. Above 800 F the amount of breakdown to ammonia increased rapidly.

Morpholine, in contrast, is relatively stable at boiler pressures up to 2500 psi and steam temperatures up to 1200 F. The amount of breakdown of morpholine to ammonia increases quite slowly and uniformly up to 1200 F, the highest temperature tested.

Of all the materials tested morpholine most nearly fulfills the requirements for an ideal, volatile, alkaline material for pH control in steam-generating systems.

Discussion

A. A. BERK.[5] The National Aluminate Company has made very good use of its experimental laboratory boilers to solve plant-operating problems. Findings of general interest have been reported and published for the benefit of the entire industry, and these contributions are very much appreciated by those of us who work with boiler feedwaters. However, such reports occasionally are of a progress nature and the conclusions appear to be premature if based only on the reported data. The author of the present paper has ably reported the data from a large number of tests and the writer has no quarrel with these data, but the story is far from being complete.

The question of other decomposition products besides ammonia discloses a weakness in the author's discussion. Apparently, only condensate pH and ammonia were measured as an index of amine breakdown. This is curious because the molecular configuration of morpholine would not lead a chemist to expect ammonia to be the chief pyrolysis product. Certainly, the pH of the supposedly CO_2-free condensate from most of the tests is too low to correspond to the ammonia content alone and therefore would tend to indicate substantially complete breakdown of the amine.

The relatively low pH found in the steam condensate is attributed by the author to unusually high absorption of carbon dioxide by the stored feedwater. The author's data show that, after tenfold concentration, the boiler-water methyl-orange alkalinity was 100 ppm. The feedwater alkalinity was therefore 10 ppm. Since pickup of carbon dioxide by alkaline solutions from air is slow and even maximum pickup (to bicarbonate) would be relatively small, it would appear obvious that the CO_2 content of the condensate was not more than 3 or 4 ppm. Instead of being 34 ppm, the morpholine content of the steam must therefore have been very low to explain the low pH measured in the condensate. If so, the percentage breakdown of morpholine must have been very high under all of the conditions reported. For this reason, it is rather difficult to attach much significance to the results as demonstrating stability of morpholine at the indicated operating conditions.

AUTHOR'S CLOSURE

Mr. Berk has called attention to a point which needs clarification since the relationship between morpholine dosage and resultant pH in the condensate shown in this paper is not in good agreement with the results which have been obtained in plant boilers.

At the time these tests were made it was assumed that the small mixed-bed deionizing column used to remove the last traces of

[5] Chief, Industrial Water Branch, Bureau of Mines, College Park, Md.

dissolved solids was also removing all of the carbon dioxide. Later tests showed that it did not have sufficient capacity to remove all of the CO_2 present. Consequently, the supposedly carbon-dioxide-free water used in these tests actually contained relatively high amounts of CO_2 from the original water supply. As a result, most of the 34 ppm of morpholine added in the feedwater was combined with the CO_2 in the condensate and did not contribute to the measured pH increase reported.

Later experimental boiler tests at 1500 psi in which the feedwater was degassed showed that less than 5 ppm of morpholine in the feedwater is required to produce a pH of over 9.0 in the condensed steam. These later results agree much more closely with field results where only a few ppm of morpholine is required to maintain pH readings of 8.7 to 8.9 in the condensate system of high-pressure low-make-up boilers.

Mr. Berk is to be commended for his close observation in noting this discrepancy in the data presented. The benefit of his interest and experience in this subject is greatly appreciated.

Compressor Surge and Stall Propagation[1]

By H. W. EMMONS,[2] C. E. PEARSON,[3] and H. P. GRANT,[3] CAMBRIDGE, MASS.

Compressor surge is shown by the application of several types of instrumentation (notably a hot-wire anemometer) to consist of two distinct types of phenomena. The whole compressor flow system may be unstable in the manner of a self-excited Helmholtz resonator. The theory of this instability is presented and is shown to explain some of the observed pulsation symptoms. The stalling of the flow through the blade rows, which usually is assumed to be the origin of pulsation, is shown to occur in propagating groups of 1 to 5 regions involving from 2 to 20 blades each. The theory of this "stall propagation" shows the propagation velocity relative to the wheel to be dependent upon boundary-layer growth parameters and hence the frequency (relative to a stationary probe) to be proportional to the wheel speed. Another part of observed compressor pulsation thus is explained. These two phenomena frequently interact to produce complex performance characteristics. The theories presented are essentially correct as shown by experimental verification, but much remains to be done to make quantitative compressor-performance prediction practical.

Introduction

IN THE long run, the successful prediction of the performance of a bladed compressor requires a qualitative knowledge of the flow behavior likely to be encountered in the machine and adequately quantitative mathematical methods for describing the behavior in terms of accurately measurable quantities.

In a small and useful area of the possible range of design and operating parameters of a multistage axial-flow machine (closely spaced blades, low air-incidence angles, low air-turning angles, large hub-tip ratio, small inlet boundary-layer thickness), both these requirements have been met. The flow has been observed experimentally to be nearly two-dimensional (small radial flow) and steady, and the mathematics describing viscous compressible flow under these conditions has been made manageable.

In other areas, however, prediction of performance has not been noticeably successful as yet. The demand for a fuller understanding is perhaps most pressing in the high-angle-of-incidence low-flow regime of compressor operation. Here, violent over-all instabilities identified with the generic term "surge" most often have been reported, typical symptoms being audible thumping and honking at inlet and exit (at frequencies as low as 1 cycle per sec [cps]), severe mechanical vibration, and oscillation of recorded pressures throughout the machine.

Violent surge cannot be tolerated in an aircraft gas turbine since there is obvious danger of mechanical failure or interruption of the combustion process; but if the machine is to start without elaborate auxiliary equipment and if it is to operate during the sudden accelerations and rapid changes in inlet conditions encountered in highly maneuverable aircraft, then the compressor occasionally must operate at low flow rates and high incidence angles. The problem of predicting and controlling surge behavior is clearly an important one and is becoming increasingly important as size and rotational speeds of engines steadily increase.

Experimental evidence has been vague and conflicting (partly because of the limitations of conventional instrumentation). Of the surge frequencies which have been reported, some have varied with the speed of the machine and others have not. Volume of connected ducting affected the violence and frequency of surge in some reported cases and did not in others. (Examples will be given later.) That no successful analysis or prediction of surge behavior has been reported previously is not surprising, since what sketchy qualitative information was available indicated only that no single simple flow phenomenon was responsible. It seemed that there well could exist several modes of instability, and that the variety of surge symptoms reported was due to the occurrence of various combinations of them in any one machine. These ideas could remain only in the realm of speculation until more indicative experimental evidence became available.

It now can be reported that new evidence has been obtained by applying a hot-wire instrument developed at Harvard (1)[4] to a thorough study of velocity and pressure fluctuations in a centrifugal compressor with a rotating axial-flow inducer blade row, and in a single-stage axial machine. Each had shown previously its own set of surge symptoms in the low-flow operating range. The hot-wire investigations indicated that two distinct phenomena predominated, one a Helmholtz "organ-pipe" resonance in the larger chambers of the machines, and the other a particular form of blade-row flow instability characterized by the propagation of blade stall from blade to blade along the axial-flow rows in the direction of relative inlet tangential velocity. Both phenomena were present in both machines, and either could produce certain external symptoms of "surge"; between them they accounted for all observed symptoms. Phenomenologically sound theoretical explanations could be found for each, and though at present the analyses hardly could be termed adequately quantitative, criteria for the onset of surge symptoms and methods for avoiding them can be suggested, and the most hopeful direction for further investigation can be indicated.

We shall consider in Section 1 the initial experimental observations, and in Sections 2 and 3 the analyses of the two flow phenomena, together with additional experimental checks.

1 EXPERIMENTAL INVESTIGATION OF SURGE IN TWO COMPRESSOR-TEST RIGS

Introductory Remarks

During extensive conventional performance testing in the two machines to be described, operators had reported symptoms of unstable flow behavior in a variety of operating regions. This information was supplemented at an early date by some records of low-frequency wall static-pressure fluctuation at points of interest along the flow path in each machine. These observations

[1] The results presented in this paper form a portion of recent research supported by Pratt and Whitney Aircraft, to whom the authors are indebted for permission to publish the paper.

[2] Professor of Engineering Sciences, Department of Engineering Sciences and Applied Physics, Harvard University. Mem. ASME.

[3] Harvard University.

Contributed by the Gas Turbine Power Division and presented at a joint meeting of the Gas Turbine Power and Hydraulic Divisions at the Annual Meeting, New York, N. Y., November 29–December 4, 1953, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, September 9, 1952. Paper No. 53—A-65.

[4] Numbers in parentheses refer to Bibliography at end of paper.

will be examined briefly, for they are typical of the information commonly available to engineers concerned with the surge problem. The results of a detailed qualitative study of velocity fluctuations within the two machines, carried out with the hot-wire instrument of reference (1), then will be considered.

Nomenclature used in this section is as follows:

a = local speed of sound, fps
i = air incidence angle (relative to tangent to mean camber line at leading edge), deg
p_t = area-average total pressure, psf
A = area, sq ft
U = rotor-tip speed, fps
W = weight flow, lb per sec
ρ = air density, pcf
$()_1$ = condition ahead of compressor
$()_2$ = condition well behind compressor

Test Stands

The two test rigs are of widely different physical characteristics. The single-stage axial stand was designed to avoid extensive ducting in series with the compressor. Laboratory air is drawn through a short annular elbow directly into a test section containing rotor and stator, and exhausted immediately back into the room through a movable cone throttling diffuser.

The centrifugal test stand was designed to throttle at either inlet or exit and to discharge outside the building; these features required a large chamber containing calming screens and straighteners between the inlet throttling (butterfly) valves and the test section, and some 40 ft of ducting and valves at the exit. The test section itself contains an axial-flow inducer-blade row and a straight-radial-bladed centrifugal impeller followed by a vaneless radial diffuser and collector case.

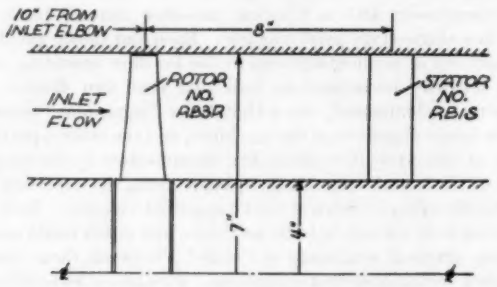


Fig. 1 Axial-Flow Compressor Stage

Several stages were tested in the single-stage axial machine and all showed similar surge symptoms. The rotor-stator combination in Fig. 1 was studied in greatest detail. It has characteristics typical of early stages in a multirow axial-flow machine, i.e., fairly low hub-tip ratio, high gap-chord ratio (1.7 at mean radius), and moderate turning (rotor-blade camber angle 30 deg at mean radius).

All testing in the centrifugal stand was carried out on the single inducer-impeller-diffuser combination shown in Fig. 2 whose principal physical characteristics are low inducer hub-tip ratio, large inducer camber angle (60 deg), and moderate impeller hub-tip ratio.[5]

Over-All Performance Tests

Curves of total-pressure ratio versus flow for each machine are shown in Figs. 3 and 4. This information was recorded during conventional survey programs in which flow was varied by exit throttling at each of a series of rotor speeds in the low-speed range where violence of surge was not excessive. Instrumentation consisted of arrays of fixed probes well ahead of and well behind the compressors. Flow was metered by a previous calibration of inlet-passage-wall static taps in each machine.

It can be observed from Figs. 3 and 4 that a variety of pressure-rise-curve shapes were encountered. The axial compressor yielded

[5] The inducer and impeller of this centrifugal compressor were matched badly. It should be mentioned that both the axial and centrifugal test stages were old experimental designs.

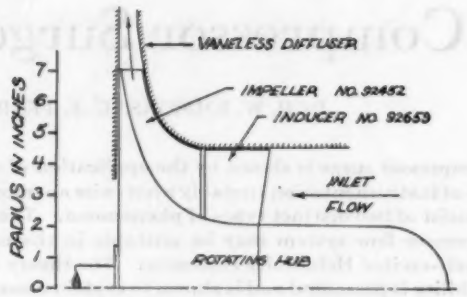


Fig. 2 Centrifugal Test Rotor

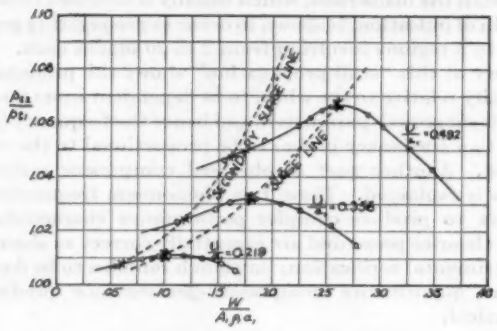


Fig. 3 Axial Compressor Pressure Ratio Versus Mass-Flow Factor

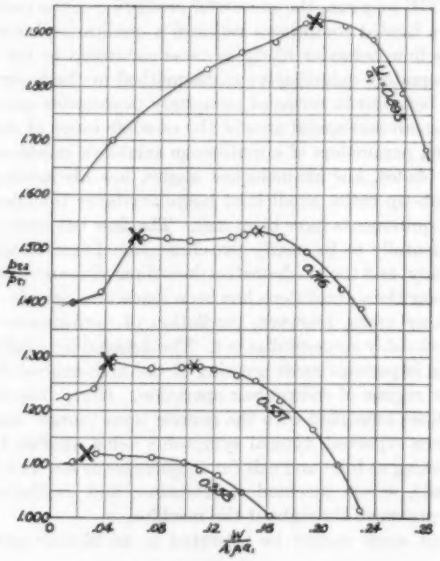


Fig. 4 Centrifugal Compressor Pressure Ratio Versus Mass-Flow Factor

smoothly humped curves at all speeds, while the centrifugal machine produced characteristics of three distinct shapes, including (*a*) at lowest speeds, a smoothly rising pressure ratio from maximum flow to nearly zero flow, with sudden drop there, (*b*) at high speed, a double peaked characteristic, (*c*) at high speeds, a curve peaking near maximum flow and falling smoothly between that point and zero flow.

Surge points reported at various times are marked with heavy crosses in Figs. 3 and 4. In the axial-stand tests, all operators agreed that noise level increased sharply and recorded pressures tended to become unstable at the peak of the performance curve at each speed, although no predominant frequency of audible noise was reported. The locus of these points was designated the surge line of the compressor. It was noticed that at a somewhat lower flow rate at each rotor speed, low frequencies (less than 100 cps) that varied with rotor speed became distinct, and pressures read at upstream wall static taps rose slightly, suggesting local flow redistribution. The locus of these points was designated the "secondary surge line." As the exit throttle was closed further, noise level diminished steadily, until at zero flow operation was quiet and, to all appearances, stable. Upon reopening the exit valve the sequence was repeated in reverse, with a slight hysteresis behavior in static pressures reported in the vicinity of the secondary surge line.[6]

In the centrifugal stand tests, the reports of surge behavior were particularly confusing; as flow was decreased at any fixed rotor speed, a point eventually was reached at which a violent thumping (at about 10 cps at all speeds) became evident. Mechanical vibration became severe and pressures read at upstream wall static taps rose well above inlet total pressure. At the lower speeds this point coincided with the sharp drop in pressure ratio near zero flow; at higher speeds it occurred with the gradual drop in total pressure ratio from the peak value near maximum flow. In addition, at intermediate speeds, some operators reported sharp increases in noise level at the points indicated by the lighter crosses, although the 10-cps frequency was not distinctly audible.

Pressure-Fluctuation Observations

Measurements of low-frequency pressure fluctuations were made at various stations along the casing of each machine by the simple hot-wire technique diagrammed in Fig. 5. A hot-wire

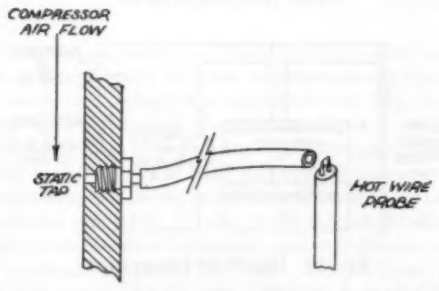


Fig. 5 Instrumentation for Low-Frequency Pressure-Fluctuation Measurements

probe (of the type shown later in Fig. 9) is held at the open end of a length of plastic tubing leading from a wall static tap. Static-pressure fluctuations show up as oscillations in the velocity of air of the tubing; the resultant fluctuations in the voltage across the wire (operated at constant heating current) are amplified and presented on an oscilloscope for examination. High frequencies are damped out by the use of a 10-ft piece of tubing. This instrument provides a good quantitative measure of surge frequency and a qualitative picture of violence of surge.

The results of these surveys are shown in Figs. 6 and 7. When flow is reduced at constant rotor speed corresponding to U/a_1 =

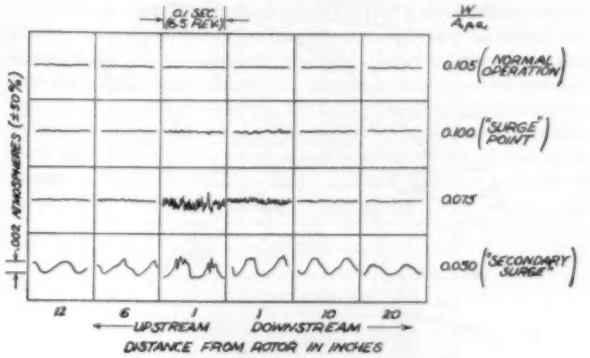


Fig. 6 Pressure Fluctuations in the Axial Test Stand; $U/a_1 = 0.219$

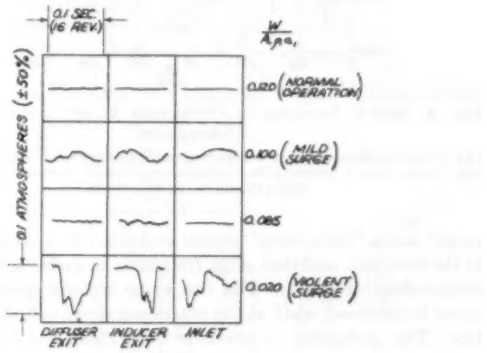


Fig. 7 Pressure Fluctuations in Centrifugal Test Stand; $U/a_1 = 0.537$

0.219 in the axial compressor, Fig. 6, the flow becomes noticeably rough in the vicinity of the rotor as the surge line is crossed, although flow a few inches either side of the rotor remains fairly smooth. Shortly before the secondary surge line is reached, a large 200-cps pressure fluctuation becomes predominant in the vicinity of the rotor; pressure fluctuations elsewhere are still negligible. At the secondary surge line, an enormous 20-cps pressure pulse appears abruptly throughout the machine, with maximum amplitude near the rotor, but dying in amplitude only slightly upstream and downstream. Similar tests at other rotor speeds with this stage showed identical behavior, except that the frequencies encountered just before secondary surge and in secondary surge varied almost directly with rotor speed. This behavior defied analysis; clearly the acoustic dimensions of the machine were not the controlling factor, and thus the most convenient explanation of surge behavior was eliminated.

In the centrifugal stand, Fig. 7, a similar set of measurements revealed a different behavior. It became evident that at the rotor speed tested there existed two clearly defined surge regions (in which low-frequency pulsing occurred throughout the machine) separated by a region of stable operation (in the sense that no low-frequency pulsing was observed anywhere in the machine). In the first surge region the predominant frequency was 10.5 cps;

[6] When the exit valve was opened very carefully from the completely closed position, steady over-all backflow through the entire machine sometimes could be obtained.

in the second, more violent, surge region near zero flow, the predominant frequency was 9.5 cps. The series of inlet-pressure fluctuation tests, summarized in Fig. 8, were run subsequently to define pulsating-flow regions throughout the operating range of this test stand. At low rotor speeds, note that there is a "mild

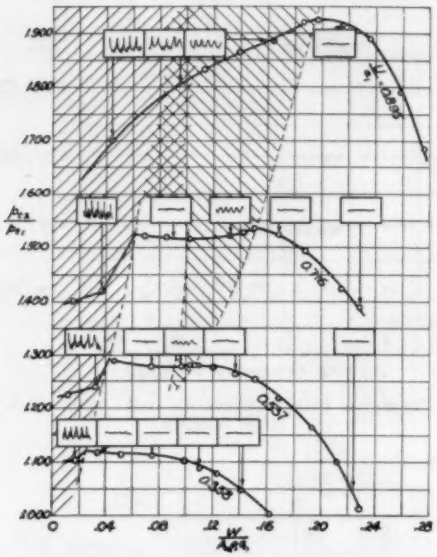


FIG. 8 SURGE REGIONS IN OPERATING RANGE OF CENTRIFUGAL COMPRESSOR

(As shown by inlet pressure-fluctuation oscillograms; each oscillogram shows one 1/2-sec sweep; pressure fluctuations are roughly to scale; shading indicates extent of surge regions.)

surge" and a "deep surge" region separated by a stable range, as in the first test, and that surge frequency in each region is nearly independent of speed. The two surge regions spread as rotor speed is increased, until at the maximum speed tested, they overlap. The ambiguity of previous observations is easily understood.

HOT-WIRE VELOCITY-FLUCTUATION SURVEYS

The underlying causes of these various compressor-surge symptoms became clear with the application of the constant-current hot-wire instrument developed at Harvard (1). The hot-wire principle is familiar. The electrical resistance of a metal varies with its temperature; therefore, when a tiny electrically heated tungsten wire (operating at constant heating current) is exposed to a cooling-air stream of known temperature, the average voltage across the wire becomes a measure of the component of average stream velocity which is perpendicular to the wire (that component being, in general, the only one effective in cooling a thin wire). It was shown (1) that when the heated wire is exposed to large and rapid velocity fluctuations, the magnitude of the instantaneous time derivative of the resulting fluctuating voltage across the wire is a direct measure of the instantaneous velocity perpendicular to the wire.

For precise measurement of instantaneous velocity, a careful calibration of each wire is required,[7] and a fairly accurate record of air temperature and "cold-wire" resistance during subsequent tests must be kept. For qualitative work, simplification is possible, since over a wide range of operation the relation between instantaneous velocity and time derivative of wire voltage is nearly linear, and the constant of proportionality is affected only slightly by large changes in air temperature or even by large collections of dirt on the wire. A record of voltage time derivative versus time is obtained easily using conventional oscillographic techniques, so a rapid qualitative survey of velocity fluctuation with a precision of about ±20 per cent is relatively simple. Calibration of each wire is unnecessary; the scale factor between oscilloscope deflection and velocity fluctuation is obtained easily by placing each new wire at a standard reference point where the velocity fluctuation is known. This procedure is sufficiently accurate for the purpose of qualitative surveys, such as are presented here.

The hot-wire probe used in the present investigation is a modification of that described in reference (1), a sketch of which is shown in Fig. 9. A length of tungsten wire is supported by two tapered steel needles fixed in a lucite insert pressed into the end of a steel tube. Current-carrying leads soldered to the needles lead through the tube to a battery supply. The complete electric circuit is shown in Fig. 10; a large resistor in series with the

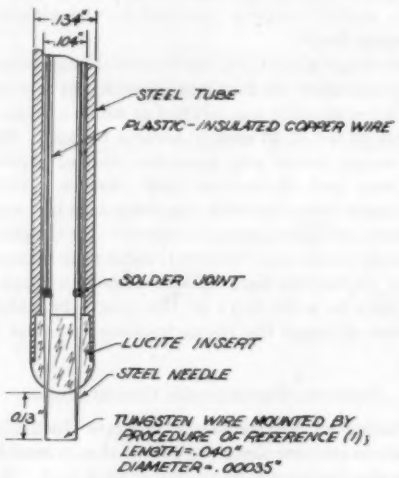


FIG. 9 HOT-WIRE PROBE

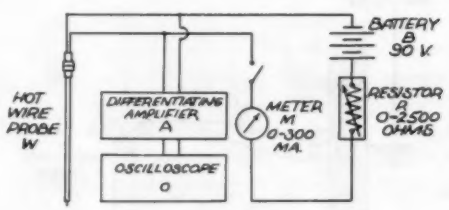


FIG. 10 HOT-WIRE INSTRUMENT

hot wire renders its current almost independent of its resistance. The derivative of the voltage across the hot wire is amplified by A, and presented on an oscilloscope screen O. As a matter of practical interest, it may be mentioned that the source of most hot-wire difficulties is the tungsten-steel connection. It is necessary that very careful copperplating and soldering techniques be employed. With experience, wires with a useful life of over 100 hr of operation may be constructed. Occasionally it is necessary to clean off the film of thick sludge which gradually accumulates on the wire because of the presence of oil in the air.

[7] A wire needs to be calibrated only once in its lifetime. The simple correction method of reference (1) compensates for changes in wire characteristics due to plastic flow, oxidation, change in steel-copper-tungsten contact resistance, etc.

Experimental Results

In general, flow behavior was examined in both machines at all points accessible to a probe. The most interesting results were obtained close behind the axial-flow blade rows in each stand. The behavior of the axial component of velocity at inducer exit in the centrifugal stand was perhaps the most spectacular and will be considered first.

In general, with the wire perpendicular to the rig axis and close behind the rotating inducer, we would expect to see velocity patterns similar to those obtained in pressure traverses behind a stationary cascade. Fig. 11 shows typical cascade exit-velocity profiles over a wide range of incidence angles and indicates the predicted shapes of differentiated hot-wire signals behind a rapidly moving cascade over a similar incidence range. The hot wire should depart from the correct velocity profile only under conditions of reverse flow (since the wire does not distinguish sense of flow). Some clipping of sharp velocity maxima and minima might be expected because of the finite length of the wire.

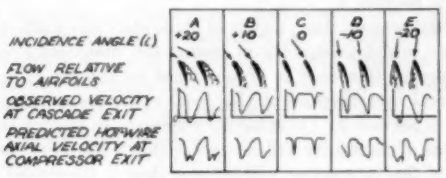


Fig. 11 Predicted Hot-Wire Response

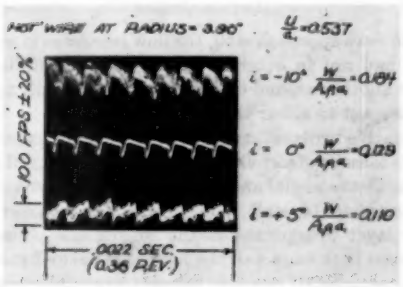
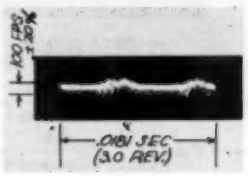


Fig. 12 Oscillograms of Axial Velocity Fluctuations at Exit of Inducer-Blade Row

Fig. 12 shows oscillograms of the inducer-exit axial velocity at a radial position midway between mid-passage and tip; the flow points correspond to incidence angles of —10 deg, 0 deg, and +5 deg, i.e., to conditions of maximum, normal, and incipient surge flows. Several traces are superposed on each oscillogram. Since time goes from left to right, and velocity increases upward, the distributions can be compared directly with those of Fig. 11. The predicted profiles B, C, and D are indeed obtained. The predicted profile E was not obtained, probably because of the inability to obtain sufficiently high flow. The predicted profile A was not observed clearly, since an instability behavior developed at about an incidence angle of 6 deg, that is, at a flow point slightly removed from the surge point. Fig. 13 shows the typical flow behavior encountered at this point; groups of stalled blades and groups of unstalled blades passed the hot wire alternately in a periodic pattern. The possibilities are now twofold; either the entire rotor stalls and unstalls periodically as a unit, or cells of stall are present on the wheel and rotate around it in time (relative to a stationary system). Since the pattern of Fig. 13 was obtained at all circumferential positions, the possibility of intermittent stall in a local region need not be considered. The frequency with which the stall cells appeared was not an integral multiple of rotor frequency, thus eliminating the possibility of stall cells fixed to the wheel.

To investigate the behavior further, the hot-wire instrument diagrammed in Fig. 14 was constructed; the existence of the two channels now permitted simultaneous measurements at various pairs of points. It was found that there were three cells of stall propagating around the wheel, each involving about three stalled blades. The direction of propagation relative to the wheel was opposite to wheel rotation, and the velocity of propagation was about 0.75 that of the wheel; that is, the cells moved past the hot wire in the same direction as that of wheel rotation with the velocity of about 0.25 that of the wheel. It also was found that these stalled cells were wedge-shaped, being widest at the tip as shown in Fig. 15. Further study revealed that the velocity of propagation varied directly with wheel speed.



Radius = 3.90 in. $\quad \frac{U}{a_1} = 0.537$

$i = +6$ deg $\quad \frac{W}{A_1\rho_1 a_1} = 0.108$

Fig. 13 Axial Velocity Fluctuations Behind Inducer-Blade Row

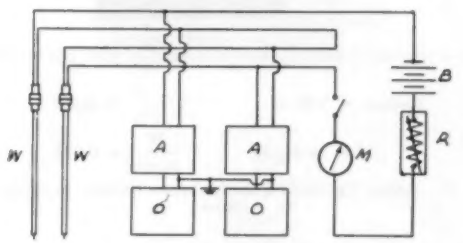
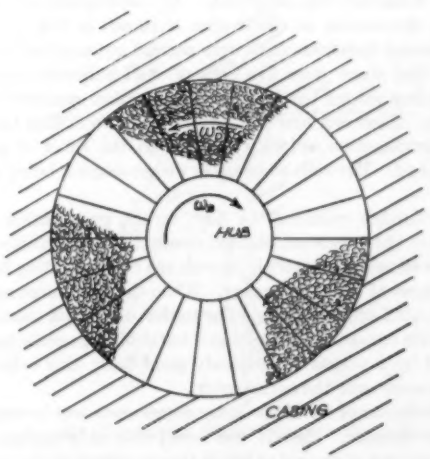


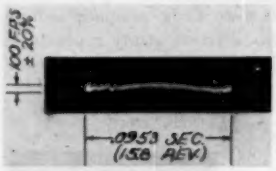
Fig. 14 Two-Channel Hot-Wire Instrument



ω_B = velocity of blades relative to casing
ω_D = velocity of stall cells relative to blades

Fig. 15 Three-Cell Stall Pattern

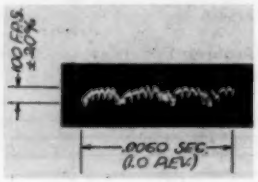
When compressor flow was reduced slightly, the mild surge region was entered and the inducer behavior of Fig. 16 was observed. Prolonged bursts of blade stall were found to occur regularly at the previously recorded surge frequency of 10.5 cps. The oscillogram pattern represents a period of 16 consecutive wheel revolutions, or 1 surge cycle. Simultaneous hot-wire pictures showed that this surging was a universal behavior, the over-all flow changing from that corresponding to an incidence angle of 0 deg into stall and back again. As previously mentioned, this surge frequency did not depend on wheel speed. Note in Fig. 16 that during that part of the cycle in which flow is passing from good to bad, stall propagation is again in evidence.



Radius = 3.90 in. $\frac{U}{a_1} = 0.537$

$i = +8$ deg $\frac{W}{A_1\rho_1 a_1} = 0.100$

FIG. 16 AXIAL VELOCITY FLUCTUATIONS BEHIND INDUCER-BLADE ROW



Radius = 3.90 in. $\frac{U}{a_1} = 0.537$

$i = +10$ deg $\frac{W}{A_1\rho_1 a_1} = 0.085$

FIG. 17 AXIAL VELOCITY FLUCTUATIONS BEHIND INDUCER-BLADE ROW

When flow was reduced beyond the mild surge point into the stable operating region between mild and deep surge, a third type of flow behavior was observed. An oscillogram of the axial-velocity fluctuation in this region is shown in Fig. 17. Again, simultaneous measurements were carried out and led to the conclusion that there were five cells of stall uniformly spaced and propagating around the wheel in the direction opposite to that of rotation. Each cell now involved about two stalled blades and had a propagation velocity relative to the rotor of about 1/4 wheel speed. The cells were again wedge-shaped, being widest at the tip.

Upon further reducing the flow, erratic propagation patterns appeared; these were supplanted eventually by complete stall, the hot wire showing large-scale turbulence corresponding to separation in most of the exit passage. When the deep surge region was entered, a 9.5-cps pattern, similar to that of Fig. 16, was observed. Each cycle consists of a prolonged burst of large-scale turbulence followed by a period of relatively good flow; this behavior was simultaneous over the entire rotor.

The behavior of the single-stage compressor was investigated in a similar manner. Here it was found that only random stalling and unstalling of the rotor-blade tips occurred at the surge line. Also, it was found that a five-cell rotor-tip stall-propagation pattern rotating in a direction opposite to that of the wheel accounted for the predominant frequency previously recorded at lower flows. It was discovered, too, that in the so-called secondary-surge region rotor and stator were stall-propagating in unison, the pattern in this case being a single cell of stall extending about halfway around the periphery of the machine and extending from root to tip. This single cell also rotated in a direction opposite to that of the wheel; its frequency of appearance at a fixed position on the periphery was that of the previously observed secondary surge. The direct variation of this surge frequency with wheel speed was simply a manifestation of the dependency of propagation velocity on wheel speed. Then in fact there was no true over-all surge in the axial compressor studied. (This result should not be generalized since such surge might be expected in stages designed for more work.)

It thus appears that two distinct flow phenomena are responsible for observed compressor surge properties:

1 Helmholtz resonance type of system instability whose frequency is expected to be complexly dependent upon the characteristics of the compressor and its inlet and exit duct connections. A theoretical investigation of this over-all surge is presented in Section 3.

2 Stall propagation; a type of blade stall which affects groups of blades, which groups travel around the compressor row. The frequency in this case is proportional to the wheel speed. A theoretical investigation of this local stall type of surge is presented in Section 2. (It is hardly necessary to remark that these two types of instability will interact with one another; the general nature of this interaction will become clear in the sequel.)

2 STALL PROPAGATION

Introduction

As blade rows approach stall, the flow separates in some groups of blades and not in others. Such separated (stalled) regions follow one another around the wheel and are usually not stationary with respect to either the wheel or the case.

A qualitative explanation of this phenomenon is easy to give. Consider a cascade of airfoils operating on the verge of stall (large positive incidence angle) and suppose some transient disturbance causes one vane to stall momentarily (i.e., causes the vane boundary layer to separate on the convex side); then the flow area between that vane and the next is restricted momentarily. Since the cascade-exit pressure remains essentially constant (dependent on the mean turning through the whole blade row) the local flow must change, and, in fact, the result is a momentary choking of flow through the single blade passage. Flow ahead of the blade passage is diverted toward adjacent passages. The effect is to increase the angle of attack on the vane next in line in the direction of tangential component of inlet velocity and to decrease the angle of attack on the preceding vane, Fig. 18. Thus

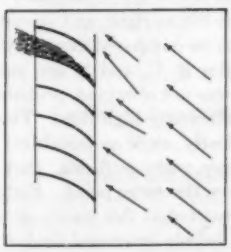


FIG. 18 EFFECT OF STALLED VANE ON INLET FLOW

the next vane tends to stall and the preceding vane tends to become more stable. If the adversely affected vane does stall, it in turn affects the one next to it, and in this manner stall spreads along the blade row in the direction of inlet tangential velocity.

But when the stall has propagated some distance, the incidence angle on the first vane improves, because of the flow being diverted by the choking of succeeding blade passages, so that the first vane unstalls, and the unstalling process subsequently moves from blade to blade down the row. One might expect to find stall regions of a characteristic size propagating down the blade row with a velocity which depends upon the rate of formation and recovery of the separation regions.

The quantitative formulation of the problem involves several steps. By a small disturbance analysis, the condition for the appearance of large-scale disturbances can be obtained, and this is done. Following this, another analysis is required to determine the mean row characteristics observed during stall propagation; such an analysis would treat finite disturbances and would lead to the size of the separation region as well as to the resultant mean characteristics. Such a subsequent analysis is as yet not sufficiently useful to justify the labor involved.

Preliminary Definitions

Since the picture of the cause of stall propagation involves the changes of flow through a cascade under conditions of varying angle of attack, it is well to recast partly the customary cascade characteristics in a more appropriate form. The total-head profiles obtained behind a blade row usually are interpreted with the aid of a loss coefficient ζ defined by

$$\zeta = \frac{1}{A_a}\int_{A_a} \frac{p_{10} - p_{20}}{\frac{1}{2}\rho V_1^2}\, dA \qquad [1]$$

and of a pressure coefficient C_P defined by

$$C_P = \frac{1}{A_a}\int_{A_a} \frac{p_2 - p_1}{\frac{1}{2}\rho V_1^2} \qquad [2]$$

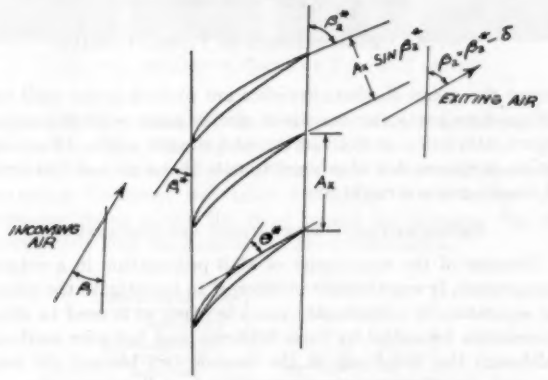


Fig. 19 Flow Through Cascade

where the notation is as follows (see Fig. 19):

A_a = exit area of one flow passage
p_{10}, p_{20} = total head before and after cascade, respectively (measured in a reference system stationary with respect to the cascade)
p_1, p_2 = static pressure before and after cascade, respectively
ρ = density, assumed constant
V_1, V_2 = velocity before and after cascade, respectively

Let us define an effective flow area A_E as that area which would suffice to carry the entire exit flow if there were no losses. Thus if W is the flow rate

$$A_E = \frac{W}{[2\rho(p_{10} - p_2)]^{1/2}} = \int_{A_a} \left(\frac{p_{20} - p_2}{p_{10} - p_2}\right)^{1/2} \sin(\beta_2^* - \delta) \qquad [3]$$

where β_2^* is the angle of the trailing edges and δ the air-stream deviation, as shown in Fig. 19. Define next a flow coefficient α by

$$\alpha = \frac{A_E}{A_a \sin \beta_2^*} \qquad [4]$$

This quantity α will play a predominant role in the subsequent calculations; since most cascade correlations yield only ζ and C_P an approximate formula (assuming $\delta = 0$) relating α to these two quantities now will be derived.

Let $dx = dA/A_a$, so that x ranges from 0 to 1, and assume that the exit total pressure is that corresponding to a parabolic loss region of fractional extent y, i.e.

$$p_{20} = p_2 + \left(\frac{x}{y}\right)^2 (p_{10} - p_2) \text{ for } 0 < x < y$$

$$p_{20} = p_{10} \text{ for } y < x < 1$$

Then the equations for α and ζ yield

$$\alpha = 1 - \frac{1}{2}y, \quad \zeta = (1 - C_P)\frac{4y}{3},$$

whence

$$\alpha = 1 - \frac{3}{8}\left(\frac{\zeta}{1 - C_P}\right) \qquad [5]$$

Small Disturbance Analysis

Consider the flow of an incompressible fluid through a cascade

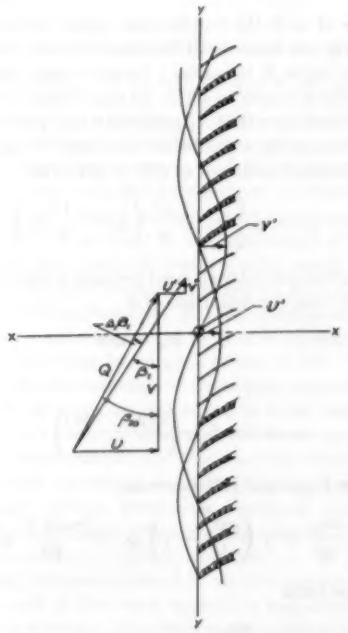


Fig. 20 Disturbed Flow Through Cascade

with an inlet velocity (far upstream) of magnitude Q and angle β_{10}. Fig. 20 shows the arrangement. The steady operation of the cascade will correspond to an inlet-flow angle of β_{10} and to a

flow-area coefficient of α_{00}. Suppose now that the flow is disturbed so that additional flow passes through certain sections and less through others. The disturbance velocities u' and v' are the derivatives of a disturbance velocity potential ϕ

$$\left.\begin{aligned} u' &= \phi_x \\ v' &= \phi_y \end{aligned}\right\} \phi_{xx} + \phi_{yy} = 0 \quad \ldots\ldots [6]$$

Separation of variables and use of the condition that $\phi \to 0$ as $x \to -\infty$ gives

$$\phi = \sum_1^\infty \left(a_n(t) \cos \frac{n\pi y}{b} + b_n(t) \sin \frac{n\pi y}{b} \right) e^{\frac{n\pi x}{b}} \quad \ldots\ldots [7]$$

where it is assumed that the disturbance is periodic in y. The values of $a_n(0)$ and $b_n(0)$ will depend on the initial shape of the disturbance; the manner in which a_n and b_n depend on t then will determine whether the disturbance grows or decays in time. The time dependence of the Fourier coefficients depends upon the cascade regarded as a boundary condition on the inlet flow. From Fig. 20

$$\tan \beta_1 = \frac{U + \phi_x}{V + \phi_y} \quad \ldots\ldots [8]$$

For purposes of stability analysis it is sufficient to assume a linear dependence between α_0 and $\tan \beta_1$, thus

$$\begin{aligned} \alpha_0 &= \alpha_{00} + \left(\frac{d\alpha_0}{d(\tan \beta_1)}\bigg|_{\beta_1 = \beta_{10}} \right) \frac{\phi_x - \frac{U}{V}\phi_y}{V} \\ &= \alpha_{00} + \alpha' \frac{\phi_x - \frac{U}{V}\phi_y}{V} \quad \ldots\ldots [9] \end{aligned}$$

This value of α is the equilibrium value corresponding to u' and v' but it is not necessarily the area actually available at the time the flow angle β_1 is reached, because some time is required for the separation region to attain its equilibrium value. Making the simple assumption that A_x grows at a rate proportional to the difference between the equilibrium area and the actual area, the effective flow-area coefficient α will be given by

$$\frac{\partial \alpha}{\partial t} = \gamma(\alpha_0 - \alpha) = \gamma \left[\alpha_{00} + \frac{\alpha'}{V} \left(\phi_x - \frac{U}{V}\phi_y \right) - \alpha \right] \quad \ldots [10]$$

where the partial derivative is used because α depends on y as well as on t. But from the definition of α

$$U + \phi_x = \omega\alpha$$

where

$$\omega = \sin \beta_2^* \cdot \left(\frac{2(p_{10} - p_2)}{\rho} \right)^{1/2}$$

and therefore Equation [10] becomes

$$\frac{\partial \phi_x}{\partial t} = \gamma \left(\frac{\omega\alpha'}{V} - 1 \right) \phi_x - \frac{\gamma\omega\alpha' U}{V^2} \phi_y$$

which has the form

$$\frac{\partial \phi_x}{\partial t} = B\phi_x - C\phi_y \quad \ldots\ldots [11]$$

This equation is the relation which must hold between the disturbance velocity components (ϕ_x, ϕ_y) at the cascade. It is essentially a boundary condition on the inlet-velocity distribution and serves to determine the Fourier coefficients of the inlet-disturbance velocities. Substitution of Equation [7] into Equation [11] then gives

$$\begin{aligned} \frac{da_n}{dt} &= Ba_n - Cb_n \\ \frac{db_n}{dt} &= Bb_n + Ca_n \quad \ldots\ldots [12] \end{aligned}$$

The oscillatory solution of these equations grows or is damped according as $B > 0$ or $B < 0$, respectively, i.e., for instability

$$\alpha' > \frac{\alpha_{00}}{\tan \beta_{10}} \quad \ldots\ldots [13]$$

Geometrically, the flow is stable relative to propagation as long as the tangent to an α-tan β_1 characteristic cuts the α-axis above the origin. If the tangent were to cut the α-axis below the origin, then by the foregoing linearized theory all types of disturbances would be unstable. This leads us to expect that in general the tangent would pass approximately through the origin. That this expectation is borne out by experiment is indicated in Fig. 21,

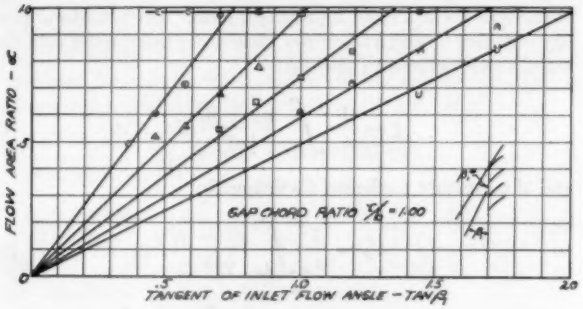


FIG. 21 CHARACTERISTICS OF TYPICAL CASCADE

where the α-tan β_1 characteristics are plotted under stall conditions for a particular cascade of camber angle = 50 deg, of gap-chord-ratio unity, and of varying inlet stagger angle. (For correlation purposes, it is of interest to note that a plot of $1/\alpha$ versus β_1 would give a straight line.)

EXPERIMENTAL OBSERVATIONS AND DISCUSSIONS

Because of the appearance of stall propagation in a rotating compressor, it was thought of interest to investigate the process of separation in a stationary cascade (such as is used to obtain correlation formulas) by both Schlieren and hot-wire methods.[8] Although the finite size of the cascade (six blades) did result in wall interference, very pronounced stall propagation was found.

Motion pictures of the stall behavior were taken, but unfortunately are not reproducible. Fig. 22 shows a sequence of Schlieren spark photographs arranged to simulate the motion-picture record. The motion pictures also disclosed an over-all surging of the mean through flow accompanying the stall propagation; this surging caused a periodic increase and decrease in the number of airfoils stalled and thereby served to obscure partially the propagation phenomenon.

At times it is difficult to determine from a Schlieren photograph whether an airfoil is stalled or merely passing to or from the stall condition. A better indication is given by a hot wire located so as to be in the wake region when a particular airfoil is stalled, but

[8] The experimental results described in this section are due to R. E. Kronauer.

in good flow when unstalled. By locating a second hot wire in a similar position at an adjacent airfoil, the velocity of stall propagation can be measured. Both this velocity and the angle of incidence at which intermittent stall first appears are functions of

TABLE 1 STALL-PROPAGATION DATA FOR CASCADE OF FIG. 22

Upstream velocity, q_1, fps	No. of stalled airfoils in a typical propagating cell	Mean incidence deg	Propagating velocity, fps	Fraction of time a typical airfoil stalled
92		14.0	3–5	
		15.0	3–5	
288	3–4	14.5	17–21	0.4
		15.0	17–21	0.6
304	2-3 1/2	14.5	13–18	0.4
		15.0	16–20	0.6
366	2-3 1/2	14.5	23–27	0.5
		15.0	29–37	0.6
		13.5	43–51	0.3
466	2 1/2-3 1/2	14.0	43–51	0.6
		14.5	40–46	0.8
566	2 1/2-3 1/2	12.5	67–75	0.5
		13.0	50–75	0.7

Cascade characteristics:

$\beta_1^* = 70$ deg $\quad \theta^* = 30$ deg

$\beta_2^* = 100$ deg $\quad r/b = 0.67$

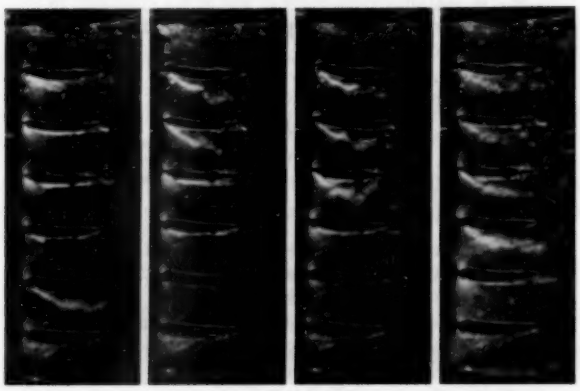


FIG. 22 SCHLIEREN VIEWS OF TYPICAL CASCADE

the mean through velocity; the results of such a measurement for the cascade of Fig. 22 are given in Table 1.

A simultaneous record of upstream and downstream-velocity fluctuation (in time) was taken by use of simultaneous hot wires, one ahead of and the other behind the cascade. Fig. 23 shows clearly that the fluctuations are synchronized.

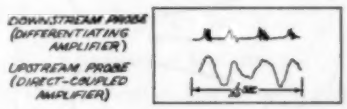


FIG. 23 SIMULTANEOUS RECORD OF UPSTREAM AND DOWNSTREAM VELOCITY FLUCTUATIONS IN CASCADE

Gain of direct-coupled amplifier is roughly 100 times gain of differentiating amplifier.)

Among the effects of stall propagation in a cascade tunnel is the excitation of the oscillation modes of the fluid in the ducting—an excitation which frequently prevents the taking of data at the bend of the loss curve.

From the solution for the velocity potential, it may be shown that the velocity of propagation of the disturbance is proportional to the wave length. This would imply that if an initial disturbance is of some arbitrary shape, this shape is not preserved during the subsequent propagation since the higher-harmonic components move more slowly. Since a fairly well-established pattern actually does occur, the finite amplitude disturbances are controlled by nonlinear effects.[9]

It may be noted that since the blade design varies spanwise in a typical blade row of an axial compressor, a portion (root or tip) of the blade will stall before the rest. The subsequent redistribution of flow then tends to inhibit stalling in the remainder of the span, thus sharpening the boundary between stalled and unstalled regions.

As the flow is decreased through a rotor, some point will attain the critical conditions for a sufficient time so that separation does occur. The diverted flow (diverted in all directions around the original spot) will cause (*a*) a sharpening of the spanwise edge of the separated spot, (*b*) a flow deterioration on the edge of the spot toward the convex side of the blades, and (*c*) a flow improvement on the edge of the spot toward the concave side of the blade. Because of the poor flow condition at the blade root and tip, one (or sometimes both) of these sections separates first. Thus propagating spots form and move around at the tip, for example, with a spanwise extension controlled primarily by the variation of cascade characteristics.

The actual flow through the compressor row, or cascade, consists of part good flow and part separated flow. The loss coefficients and pressure-rise coefficients in these two sections are different. If, however, we establish an operating point by, say, setting an inlet and exit pressure, this will determine the flow through the system and the fraction of the available area that is covered by the bad flow. While the analysis is not sufficiently complete to include in this paper it is important to note here that over-all conditions control the fraction of the area covered by separated spots while the interaction of the blades and inlet flow control their shape and propagation.

Considering the wide range of operation of a compressor, there is no hope of avoiding separation altogether. The only performance improvement that can be expected is to make the machine stall gracefully.

3 ELEMENTARY THEORY OF SURGE

Introduction and Experimental Background

In 1850, Helmholtz studied the behavior of "resonators," i.e., large jars with small necks, his assumption being that most of the kinetic energy of oscillation was to be found in the neck portion. He was able to show that the frequency of oscillation depended only on the enclosed volume and not on its shape, and that it was, in fact, inversely proportional to the square root of the volume. It was shown by later workers that a similar result held for the neck; i.e., the resonant frequency depended only on the area of the aperture and not on its shape. Various combinations of resonators and necks have been studied by people interested in acoustics; a good account may be found in reference (2).

About 1940 this phenomenon of organ-pipe resonance was used to give a qualitative picture of surging in water pumps (3) and recently essentially the same analysis was used to study nonlinear oscillations in water pumps (4). Both authors dealt with a system of reservoirs connected by narrow passageways some of which contained pumps possessing nonlinear characteristics. The analysis in each of these two cases was considerably simplified by the constant-density property of water.

When surging was encountered in air compressors, the Helmholtz theory would at first seem applicable and there is no doubt but that various attempts have been made to apply this theory to

[9] An entirely different approach, e.g., thin-wing theory, has been used by W. R. Sears. His analysis, while more complete in the inclusion of downstream effects, is less adequate in the inclusion of losses. The same propagation velocity and the same difficulties with nonlinear effects are encountered.

air-compressor surging (perhaps embodying an analysis similar to that to be discussed). However, no such analysis appears in the literature, possibly because of the fact that the predictions are apparently in error; the discrepancies disappear only when the presence of stall propagation is taken into account.

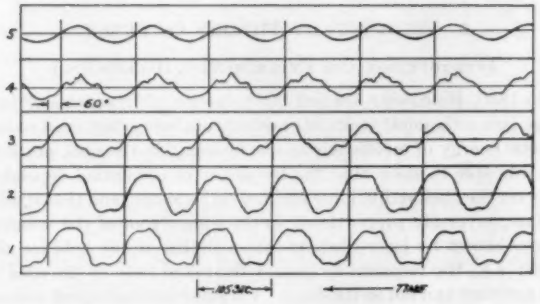
As mentioned in Part 1, it was discovered that the phenomenon previously called surge in the single-stage Harvard axial rig was not true surge, but consisted of a single-cell stall-propagation region trailing downstream from the rotor and rotating in time around the compressor axis. Because of this, it was decided to carry out all experimental surge work on the Harvard centrifugal compressor, where substantial and true surge does occur.

Two instruments were used to examine the static-pressure fluctuation in surge, the saran tubing—hot-wire arrangement of Part 1, and a steam-engine indicator-card apparatus. The former was used to measure the relative phases of the pressure fluctuations at different positions in the rig, and the latter to examine its shape and magnitude. An exhaustive series of tests were carried out, which for brevity will not be reported in detail, and the following results were obtained:

1 The maximum fluctuation in static pressure is of about the same order of magnitude as the maximum difference between inlet static and total pressure obtained in the normally operating compressor.

2 The wave form is approximately sinusoidal.

3 The fluctuations behind the rotor are almost in phase, e.g., the time-phase difference between the fluctuations at two different points behind the rotor is less than one fifth of the time required for a sound wave to pass between these two points. A similar result holds for the fluctuations ahead of the rotor; the fluctuations ahead of and behind the rotor are, however, not in phase. Fig. 24 gives a comparative survey of pressure patterns at various points in the rig, as obtained by use of a moving-film camera.



Station	Location
1	inlet chamber—60 in. ahead of rotor
2	inlet chamber—34 in. ahead of rotor
3	inlet nozzle—12 in. ahead of rotor
4	exit col. case—10 in. behind rotor
5	exit duct—60 in. behind rotor

FIG. 24 SIMULTANEOUS PRESSURE FLUCTUATIONS, CENTRIFUGAL RIG

These results imply that surging is a Helmholtz resonance effect and that it may be treated theoretically by using a linear small-oscillation theory (which always involves sinusoidal oscillations).

SURGE MODEL

Despite the simplicity of the following model, an analysis based on it turns out to predict fairly well the behavior of the centrifugal compressor.

Consider two chambers of volumes V_1 and V_2 connected to each other and to the atmosphere by means of three relatively narrow channels, the middle of which contains a compressor, as shown in Fig. 25. With reference to the figure, the nomenclature

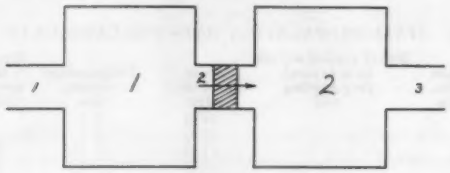


FIG. 25 SURGE MODEL

to be used is as follows:

V_i = volume of ith chamber, cu ft
p_i = instantaneous pressure in ith chamber, psf
p_0 = atmospheric pressure, psf
W_i = rate of mass flow in ith channel, lb per sec
M_i = instantaneous mass of gas in ith chamber, lb
T_i = instantaneous temperature of gas in ith chamber, deg F abs
ρ_i = M_i/V_i lb per cu ft
g = acceleration of gravity, ft/sec^2
R = gas constant = $\frac{V_i p_i}{M_i T_i}$, ft/deg = $C_p - C_v$
$P(W_2)$ = useful work per lb done by compressor (depends on W_2), ft
$P'(W_2)$ = dP/dW_2
C_p = specific heat at constant pressure, ft/deg
C_v = specific heat at constant volume, ft/deg
$\dot{\phi}$ = $d/dt(\phi)$, ϕ being any quantity
Q_i = channel constant, defined by Equation [14], ft/sec^2/lb
κ = polytropic coefficient
γ = ratio of specific heats C_p/C_V

FLOW IN A CHANNEL

Consider a channel of variable cross-sectional area, as shown in Fig. 26. Let A be the area (sq ft) at a point x ft from the left-hand end; let ρ be the instantaneous density (lb per cu ft) at this point. Then the total kinetic energy at any instant is

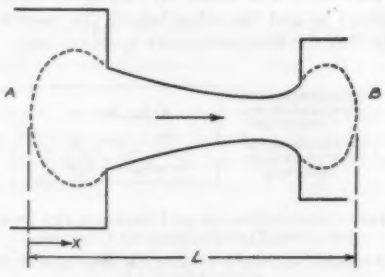


FIG. 26 CHANNEL SHAPE

$$\frac{1}{2}\int_0^L \frac{W^2}{g\rho A}\,dx$$

where W is the mass-flow rate. Assume now that because of the small size of the channel the air in it moves almost as a "unit"; then W is constant and the instantaneous kinetic energy is

$$\frac{W^2}{2}\int_0^L \frac{dx}{g\rho A}$$

Define now the "channel constant" Q by

$$Q = \int_0^L \frac{dx}{g\rho A}$$

so that the kinetic energy is

$$\text{K.E.} = \frac{1}{2} QW^2 \text{ (lb-ft)} \quad \ldots\ldots\ldots\ldots\ldots [14]$$

In calculating the changes occurring in this quantity, it may be assumed that Q is constant in time, particularly since our interest is confined to the case of initial instability (at which time the oscillations are small). In the case of a channel linking two chambers of a compressor rig, the kinetic energy of the air is not confined to the channel region but extends somewhat outward from both ends, (Fig. 26). In calculating the channel constant for such a case, it will be assumed that the integration is carried out over that region occupied by appreciable kinetic energy.

Basic Equations

If there is a compressor in the channel, then the rate of useful energy output of the compressor must equal the sum of the rate of increase of the foregoing kinetic energy and the rate at which energy is leaving the system. Using the notation of Fig. 26, where subscripts A and B refer to the conditions well ahead of and well behind the channel, i.e., to stagnation conditions, the energy equation applied to the system yields

$$WP(W) + WC_pT_A = WC_pT_B + Q\dot{W}W$$

which may be divided by W and differentiated to yield

$$P'(W)\dot{W} = C_p(\dot{T}_B - \dot{T}_A) + Q\ddot{W} \quad \ldots\ldots\ldots [15]$$

Assume now that the gas in the chamber obeys a polytropic relation of the form $p \sim \rho^\kappa$ (where κ is less than C_p/C_v because of the effect of dissipation, which is otherwise neglected). Then use of the gas law shows that

$$\dot{T} = (\kappa - 1)\frac{T}{M}\dot{M}$$

and Equation [15] then becomes

$$(P')\dot{W} = \frac{\gamma}{\gamma - 1}(\kappa - 1)R\left[\frac{T_B}{M_B}\dot{M}_B - \frac{T_A}{M_A}\dot{M}_A\right] + Q\ddot{W} \quad \ldots [16]$$

Solutions of Equations

The result of writing Equation [16] for each of the three channels in Fig. 25 is (using $\dot{M}_1 = W_1 - W_2$, etc.)

$$Q_1\ddot{W}_1 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}(W_1 - W_2) = 0$$

$$Q_2\ddot{W}_2 - P'\dot{W}_2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\left[\frac{T_2}{M_2}(W_2 - W_3) - \frac{T_1}{M_1}(W_1 - W_2)\right] = 0$$

$$Q_3\ddot{W}_3 - \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}(W_2 - W_3) = 0 \quad \ldots\ldots [17]$$

For small oscillations, P' may be assumed constant. It is a mathematical theorem that the solution of any such set of equations must have the form

$$W_i = C_i e^{\alpha t} \quad \ldots\ldots\ldots\ldots\ldots [18]$$

(or, more generally, the form of a polynomial in t multiplying $e^{\alpha t}$; such more general solutions exist only if the determinantal equation which shortly appears has multiple roots) and, therefore, we may substitute directly from Equation [18] into [17] to give

$$C_1\left[Q_1\alpha^2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right] + C_2\left[-\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right] + C_3(0) = 0$$

$$C_1\left[-\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right] + C_2\left[Q_2\alpha^2 - P'\alpha + \gamma\frac{\kappa - 1}{\gamma - 1}R\left(\frac{T_2}{M_2} + \frac{T_1}{M_1}\right)\right] + C_3\left[-\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right] = 0$$

$$C_1(0) + C_2\left[-\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right] + C_3\left[Q_3\alpha^2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right] = 0$$

For this set of equations to yield a set of nonsimultaneously vanishing C_i the determinant of the coefficients must vanish; after some simplification, this condition gives

$$\begin{vmatrix} Q_1\alpha^2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1} & Q_1\alpha & 0 \\ -\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1} & Q_2\alpha - P' & -\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2} \\ 0 & Q_3\alpha & Q_3\alpha^2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2} \end{vmatrix} = 0 \quad \ldots [19]$$

Now each root α of Equation [19] possesses, in general, a real and an imaginary part; if the real part is negative, the oscillation whose frequency is determined by the imaginary part will decrease in time, but if the real part is positive the oscillation will increase in time. The critical condition for instability, therefore, may be obtained by calculating that value of P' for which α has a zero real part. Setting $\alpha = i\omega$ and expanding Equation [19], we obtain a complex equation whose real and imaginary parts must vanish separately

$$\left(-\omega^2Q_1 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right)\left(-\omega^2Q_3 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right)P' = 0 \quad \ldots\ldots [20]$$

$$\left(-\omega^2Q_1 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right)\left(-\omega^2Q_2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right)Q_3 + \left(\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right)\left(-\omega^2Q_2 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right)Q_1 + \left(\gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_2}{M_2}\right)\left(-\omega^2Q_1 + \gamma\frac{\kappa - 1}{\gamma - 1}R\frac{T_1}{M_1}\right)Q_2 = 0 \quad \ldots\ldots [21]$$

There are three factors in Equation [20]; if either of the first two factors vanishes, Equation [21], in general, cannot be satisfied. Hence $P' = 0$ is a requisite condition for (critical) instability. Equation [21] then determines the frequency. It is solved easily since it is a quadratic in ω^2

$$\left(\omega^2 \frac{\gamma-1}{\gamma(\kappa-1)}\right)^2 + \omega^2 \frac{\gamma-1}{\gamma(\kappa-1)}\left[-R\left(\frac{T_2}{Q_2M_2}+\frac{T_1}{Q_1M_1}+\frac{T_1}{Q_2M_1}+\frac{T_2}{Q_2M_2}\right)\right] + R^2\frac{T_1}{M_1}\frac{T_2}{M_2}\left(\frac{Q_1+Q_2+Q_3}{Q_1Q_2Q_3}\right) = 0 \ldots [22]$$

To obtain an idea as to the order of magnitude of the various quantities involved, the following experimental results were obtained:

$Q_1 = 3.5$ sq ft per lb	$V_1 = 54$ cu ft
$Q_2 = 1.6$ sq ft per lb	$V_2 = 4.4$ cu ft
$Q_3 = 15$ sq ft per lb	$T_1 = 525$ F abs
$p_1 \cong p_0$	$T_2 = 550$ F abs
$p_2 \cong 1.2p_0$	Surge $f \cong 24$ cps

Then $RT_1/M_1 = 7500$, and $RT_2/M_2 = 82{,}500$; substitution into Equation [22] gives

$$\left(\omega^2 \frac{\gamma-1}{\gamma(\kappa-1)}\right)^2 - \omega^2 \frac{\gamma-1}{\gamma(\kappa-1)}\,[5400 + 2100 + 4700 + 51{,}500] + 147 \times 10^6 = 0 \ldots [23]$$

whence

$$f = \sqrt{\gamma \frac{\kappa-1}{\gamma-1}}\,(39.5) \text{ cps}$$

The value of κ is not known, but a good guess would be about 1.3. Then $f = 40$ cps; this is of the right order of magnitude when compared with the measured cycles per second but is somewhat too large. This is not unexpected, for the calculated value of each Q_i assumed that the kinetic energy was confined to the channel itself, whereas in practice it must extend considerably on both sides. This correction would decrease the calculated frequency, which is in the right direction. A much better check on the theory may be obtained by varying the exit volume. It will be noticed in Equation [23] that ω is dominated by the term $(RT_2)/(Q_2M_2)$; we therefore expect that if M_2 alone is varied, $\omega \sim 1/\sqrt{V_2}$. Experimentally, it was found that for $V_1 = \infty$, $V_2 = 4.4$ cu ft, f was 17.5 cps; for $V_1 = \infty$, $V_2 = 13.4$ cu ft, f was 9.5 cps. The two ratios are

$$\sqrt{\frac{4.4}{13.4}} = 0.57 \qquad \frac{9.5}{17.5} = 0.54$$

which agree rather well.

Knowing now the approximate relative magnitude of the quantities involved in Equation [22], an approximate expression for the frequency of pulsation may be obtained

$$f = \frac{1}{2\pi}\sqrt{\frac{\gamma(\kappa-1)}{\gamma-1}}\sqrt{\frac{RT_1}{M_1}\left(\frac{1}{Q_1}+\frac{1}{Q_2}\right)+\frac{RT_2}{M_2}\left(\frac{1}{Q_2}+\frac{1}{Q_3}\right)} \ldots [24]$$

In an actual jet engine, volume 2 in Fig. 25 would represent the volume of the combustion chambers and the volume of the tubing between these and the compressor; volume 1 would represent the volume between the inlet-duct channel 1 and the compressor. The volume and shape of channel 2 would be that of the compressor passage. In general, the entrance to the compressor is practically atmospheric; this condition is equivalent to setting $M_1 = \infty$, and Equation [24] becomes

$$f = \frac{1}{2\pi}\sqrt{\gamma\frac{\kappa-1}{\gamma-1}\frac{RT_2}{M_2}\left(\frac{1}{Q_2}+\frac{1}{Q_3}\right)} \ldots [25]$$

General Comments

Properties of Solution. It is essential, in using the model, that the entire impeller be considered as a unit. For example, if it were divided into its inducer-impeller components with a small intermediate volume, then the analysis would be erroneous because of the illegitimacy of the Helmholtz resonance assumption for so small a volume. For the same reason, all stages in a multistage compressor must be considered as lumped together if it is wished to apply the foregoing results. It is probable that this consideration is approximately valid in so far as a crude picture of over-all surge is concerned.

Because of the fact that $\alpha = 0$ is a root of Equation [19], another possible solution is $W_i = C_i' e^{0t} = C_i'$ (and by Equation [17] these C_i' must all be equal), and the general solution then is obtained by adding this constant solution to the previous oscillatory solution. In other words, the solution is valid if there is a steady air flow present, as of course there always is in practice. It also may be noted that if $P' = 0$, then $\alpha = 0$ is a multiple root, which corresponds to the case of a changing air flow. Thus, even in acceleration or deceleration, the condition $P' = 0$ must be satisfied for surging to occur, provided that the processes accompanying the change in speed are of a quasi-static nature.

The way in which the value of $(RT_2)/(Q_2M_2)$ dominates the frequency is probably worth emphasizing. Since changes in exit or entrance nozzle do not affect the value of this term, such changes would not be expected to alter the surge frequency by very much. Experiments in which a great many types of inlet and exit conditions were used (circular, elliptical, and multiple holes, bellmouth holes, pipes, and so on) bore out this expectation remarkably well.

The speed of the impeller does not enter directly into the expression for ω; it is, however, implicitly present in that the throttling condition for incipient pulsation will depend on the speed. But according to the previous remark, the change in throttling will have little effect and the change in frequency will depend on $(T_2/M_2)^{1/2}$ which would change but little, and hence it is expected that the frequency is relatively independent of speed. For small speed changes this is indeed the case.

Harvard Centrifugal and Axial Rigs. It should be noted that the condition for surge is that the slope of the useful-work-done per pound curve should vanish, not that the slope of the pressure-ratio curve should vanish. These two curves may be expected to have about the same shape for a moderately efficient compressor.

For the present centrifugal rig, the pressure-ratio curve shown in Fig. 27 indicates that the compressor should surge at point A.

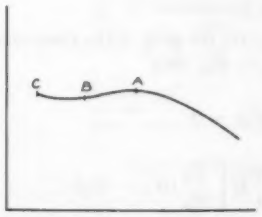


Fig. 27 Typical Centrifugal Characteristic

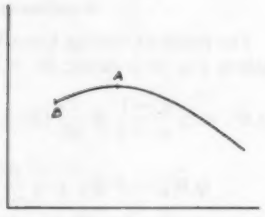


Fig. 28 Typical Axial Characteristic

This is indeed the case; further, because of the relatively gentle fall-off of the pressure-ratio curve, the pulsation is quite mild and is hardly detectable without instruments. When region B is reached, the operation is again stable; at point C the compressor drops into deep surge.

The pressure-ratio curve for the single-stage axial rig of Part 1

is reproduced in Fig. 28. Again, surge may be expected at point *A*, and, in fact, there is a sudden jump in the noise level at this point, corresponding probably to a rather high-frequency surge. The frequency of this surge may not be determined by the previous theory, because the dimensions are such that the frequency as calculated from Equation [22] is too high for the assumptions to remain valid. The theory, however, still may be expected to predict the point of incipience, as indeed it does, for the equations show that an instability of the Helmholtz type is possible, although once this instability occurs the dimensions are such as to force the resonance into a new pattern. As the flow is decreased, violent stall propagation sets in and dominates the rather weak surging.

BIBLIOGRAPHY

1 "Measurement of Instantaneous Vector Air Velocity by Hot-Wire Methods," by C. E. Pearson, *Journal of the Aeronautical Sciences*, vol. 19, February, 1952, pp. 73–82.

2 "Theory of Sound," by J. W. S. Rayleigh, Dover Publications, New York, N. Y., vol. 2, 1945, chapter 16.

3 "Untersuchungen an einer Kreiselpumpe mit labiler Kennlinie," by R. Dziallas, VDI-Verlag, Berlin, Germany, 1940.

4 "Stability and Surging of a Centrifugal Water Pump," by Sumiji Fujii, Transactions of the Society of Mechanical Engineers, Japan, vol. 13, no. 44, May, 1947, 1st report, pp. 184–191; 2nd report, pp. 192–201; and vol. 14, no. 48, 3rd report, pp. 12–17; 4th report, pp. 17–25. (in Japanese.)

5 "Compressor Surge Investigated by NACA," by R. O. Bullock and H. B. Finger, *SAE Journal*, vol. 59, September, 1951, pp. 42–45.

Discussion

W. R. Hawthorne,[10] M. D. Wood,[10] and J. H. Horlock.[10] Similar investigations to those described in the paper have been undertaken at the Cambridge University Engineering Laboratory, on an axial-flow compressor comprising three blade rows—guide vanes, rotor, and stator. The axial spacing of these rows may be varied.

These investigations, made with a hot-wire apparatus designed and built at Cambridge, have shown the existence of stalled cells of air rotating with the moving blades but with a slower angular velocity, confirming the results given by the authors.

The number and shape of these stalled patches vary not only with the mass-flow rate through the compressor but also with the axial spacing of the blade rows.

Completely different plots of total temperature rise and total pressure rise against mass flow are obtained with different axial spacing, and different stall patterns are observed. In all the compressor arrangements tested so far, the onset of stall-cell propagation on the rotating row is associated with a discontinuity in the total temperature rise-mass flow characteristic, whereas, as expected, initiation of propagation in a stator row has not been found to produce this effect.

It also has been noted that the start of rotating stall on the rotor tip may result in the unstalling of an adjacent downstream stator root. This is the reason for the differences in the over-all characteristics.

These results suggest that the effect of upstream and downstream adjacent blade rows must be included in any analysis if correct prediction of the velocity of propagation and dimensions of the stall cells in compressors are to be made theoretically.

M. C. Huppert.[11] This paper presents an interesting and timely account of surge and stall propagation. The authors were probably the first investigators to distinguish between surge and flow fluctuations due to stall propagation.

The writer's comments on the paper will serve largely to supplement the material presented. With regard to the analysis of surge, the authors neglected to include a throttle in the flow system for controlling the air flow. The inclusion of the compressor-discharge throttle complicates the criteria for stability so that $P' = 0$ is no longer the requisite condition for critical instability. In order to demonstrate the influence of the throttle, it will be convenient to resort to an electrical analogy of the flow system (see Olson[12] for a discussion of analogies between acoustical and electrical systems).

Set

$$C_i = \frac{\gamma - 1}{\gamma(\kappa - 1)} \frac{M_i}{RT_i} \text{ (capacitance)}$$

$$L_i = Q_i \text{ (inductance)}$$

$$I_i = W_i \text{ (current)}$$

$$E(I_2) = P(W_2) \text{ (electromotive force)}$$

For our purpose here, we will take

$$P(W_2) = \int_{p_{i_1}}^{p_{i_2}} \frac{dp_i}{\rho l} \cong \frac{p_{i_2} - p_{i_1}}{\rho l_{avg}}$$

An equivalent electric current for the system considered by the authors but with a throttle added is shown in Fig. 29, herewith.

The circuit element $E(I_3)$ represents the throttle and for steady flow the following conditions apply

$$I_1 = I_2 = I_3$$

$$E(I_2) = E(I_3)$$

By applying Kirchhoff's law to each of the three circuit loops shown in Fig. 29 and differentiating the resulting equations once with respect to time, the following expressions result

$$L_1 \dot{I}_1 + \frac{1}{C_1}(I_1 - I_3) = 0$$

$$L_2 \dot{I}_2 + \frac{1}{C_2}(I_2 - I_3) + \frac{1}{C_1}(I_2 - I_1) - [E(I_2)]' \dot{I}_2 = 0$$

$$L_3 \dot{I}_3 + \frac{1}{C_2}(I_3 - I_2) + [E(I_3)]' \dot{I}_3 = 0$$

where

$$[E(I_2)]' = \frac{dE(I_2)}{dI_2} \text{ and } [E(I_3)]' = \frac{dE(I_3)}{dI_3}$$

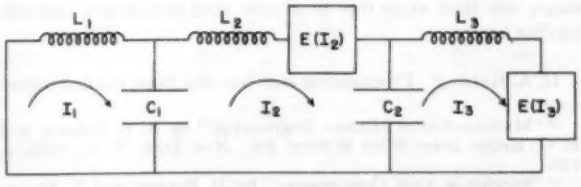


Fig. 29 Equivalent Electric Current for Surge Model

Following the same procedure used by the authors, we find an expression corresponding to Equation [19] but written as a polynomial in α as follows

[10] Cambridge University, Cambridge, England.

[11] National Advisory Committee for Aeronautics, Lewis Flight Propulsion Laboratory, Cleveland, Ohio.

[12] "Dynamical Analogies," by H. F. Olson, D. Van Nostrand Company, Inc., New York, N. Y., 1943.

$$\alpha^6 + (\tau_3 - \tau_2)\alpha^5 + (\Omega_1^2 + \Omega_2^2 + \Omega_3^2 + \Omega_4^2 - \tau_3\tau_2)\alpha^4 + [\tau_3(\Omega_1^2 + \Omega_2^2 + \Omega_3^2) - \tau_2(\Omega_4^2 + \Omega_1^2)]\alpha^3 + [\Omega_4^2\Omega_3^2 + \Omega_1^2\Omega_2^2 + \Omega_1^2\Omega_4^2 - \tau_2\tau_3\Omega_1^2]\alpha^2 + [\tau_3\Omega_1^2\Omega_2^2 - \tau_2\Omega_1^2\Omega_4^2]\alpha = 0$$

where $\tau_2 = \dfrac{[E(I_2)]'}{L_2}$ $\tau_3 = \dfrac{[E(I_3)]'}{L_3}$

$$\Omega_1^2 = \frac{1}{L_1C_1}, \quad \Omega_2^2 = \frac{1}{L_2C_2}, \quad \Omega_3^2 = \frac{1}{L_2C_1}, \quad \Omega_4^2 = \frac{1}{L_3C_2}$$

The necessary and sufficient condition for critical stability is given by Doherty and Keller,[13] as a system of relations among the coefficients in the foregoing expression.

If we consider the experimental values given by the authors, the following values are obtained

$\Omega_1^2 = 2000$ $\Omega_3^2 = 4700$
$\Omega_2^2 = 51{,}500$ $\Omega_4^2 = 5400$

The following table illustrates the effect of τ_3 on the value of τ_2 for critical stability

τ_3	τ_2 for critical stability
0	0 (authors' case)
60	5
600	10

Pearson[14] found that instabilities could occur at a compressor operating point with P' negative.

The fact that a compressor operating point is unstable does not mean that the resulting surge fluctuations will be sufficiently large to cause structural damage to the compressor. The nonlinear effects in the system will determine the amplitude of the surge fluctuations. If we assume that the major nonlinear effect is the variation of P with W_2, a distinction between two types of surge may be made. If the nonlinearity is progressive, that is, if P' changes gradually with W_2, the resulting surge fluctuations are analogous to feedback oscillations[15] and, if the nonlinearity is an essential nonlinearity such as a discontinuity in P, the resulting surge fluctuations would be analogous to relaxation oscillations. As pointed out elsewhere,[16] progressive nonlinearities in P at weight flows less than that for peak compressor-pressure ratio (where P' is positive) are probably due to a gradual or progressive stall of the compressor blading initiated by stall at one end of the blades, whereas essential nonlinearities or a discontinuity in P would likely be caused by abrupt stall.

The last reference[16] shows that a discontinuity in P is not sufficient to produce surge. The transient flow following abrupt stall must effect a stall recovery in order for surge to occur.

In general, NACA experience has been that surge resulting from progressive stall is a low-amplitude not easily detected surge, but that surge due to abrupt stall is a large-amplitude audible surge.

D. A. Rains.[17] Propagating stall has also been studied extensively at the California Institute of Technology[18] with results similar to those of the authors.

The recent construction of an axial-flow-pump test facility in the Hydrodynamics Laboratory has made possible the visual observation of stall. The pump is of compressorlike geometry and exhibits the same type of stalling phenomenon investigated by the authors. Visualization of the stall was accomplished by injecting

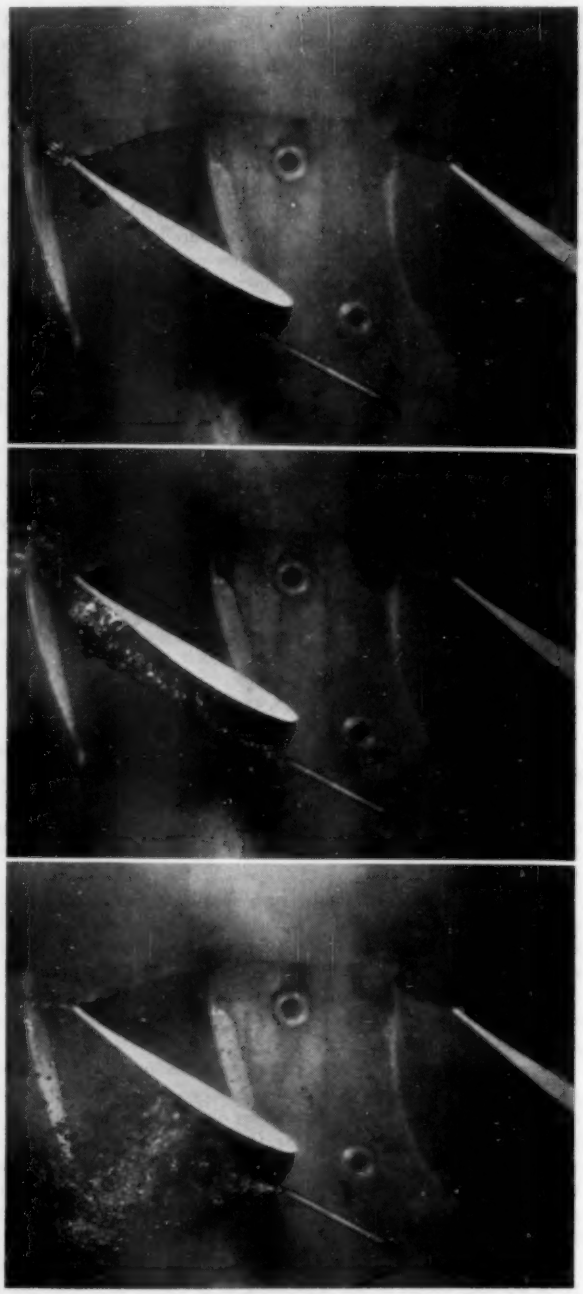


Fig. 30 Photographic Sequence Shows Development of Stall in a Passage

[13] "Mathematics of Modern Engineering," by R. E. Doherty and E. G. Keller, John Wiley & Sons, Inc., New York, N. Y., 1936, p. 129.

[14] "Surging in Axial Compressors," by H. Pearson and T. Bower, *The Aeronautical Quarterly*, published by the Royal Aeronautical Society, London, England, vol. 1, part 3, November, 1949.

[15] "Response of Physical Systems," by J. D. Trimmer, John Wiley & Sons, Inc., New York, N. Y., 1950.

[16] "Some Stall and Surge Phenomena in Axial-Flow Compressors," by M. C. Huppert and W. A. Benser, *Journal of the IAS*, December, 1953.

[17] Graduate Student, California Institute of Technology, Pasadena, Calif. Student Mem. ASME.

[18] "Experimental Investigations of Propagating Stall in Axial-Flow Compressors," by T. Iura and W. D. Rannie, Trans. ASME, vol. 76, 1954, pp. 463–471.

air through a probe attached to the rotor and observing the flow patterns with a stroboscopic light through a window in the case of the pump. The sequence of photographs shown in Fig. 30 of this discussion was selected from a number of exposures taken at random times of the same blade passage. The figure shows the development of stall in a passage as the stall pattern travels around the blade row. These photographs confirm the scheme of propagating stall as proposed by the authors, Iura and Rannie,[18] and others.

Authors' Closure

As the jet engine has become better understood at its design point, more laboratories, industrial, governmental, and academic, have turned to a more detailed study of the problem of compressor surge which exerts such major influence on the starting and part-load characteristics of the engine. No laboratory gets very far before some aerodynamic observation or blade vibration (or breaking) study discloses the phenomena of stall propagation. Thus a large number of independent discoveries have resulted.

These have given rise to a number of independent theories as to the relation between various properties of the stall propagation and the relation between this and the general compressor-surge problem. Work on the mechanism of stall as reported by the discussers is essential if we are to attain a complete understanding of compressor performance. At present none of the theories—including that of the present authors—goes far enough to make stall-spot size, propagation velocity, or resultant compressor performance predictable from a given compressor geometry.

The interaction of adjacent blade rows as reported by Hawthorne, et al., from Cambridge University is indeed important, but the mechanism of this interaction is not clear.

The remarks of Huppert relative to the effect of the nozzle characteristic are correct, although analysis by electrical analogies does not seem necessary. Following the present work, which occurred some 2 years ago, a more detailed analysis of surge has been made by one of the authors (Pearson); the resultant formulation is much more complex and considers the effects of a variety of design parameters, including throttling conditions. This work has subsequently been submitted for publication.

The method of visualization reported by Rains is very interesting and should shed considerable light on the mechanism of stall which controls the compressor properties at low flows.

Pin-Fin Heat-Exchanger Surfaces

By W. M. KAYS,[1] STANFORD, CALIF.

Plate-fin heat-transfer surfaces in which the fins are small-diameter circular pins, in contrast to fins in sheet form, are of interest in gas-flow heat-exchanger applications because of the extremely high heat-transfer conductances obtainable along with high fin effectiveness. This paper summarizes the previously available heat-transfer and flow-friction design data for such surfaces and presents new data recently obtained. A number of problems encountered in the development of these surfaces are discussed, including a serious pin-vibration problem, and a tendency for the flow to become completely laminar if the pins are in line and too closely spaced. All of the pin-fin surfaces considered have in-line pin arrangements, and it is concluded that a staggered pin arrangement will alleviate some of the undesirable features of the in-line arrangement. A heat-exchanger-design study is included to demonstrate that, despite the high friction factors characteristic of pin-fin surfaces, it is possible to design heat exchangers with these surfaces that are competitive, from a volume and weight point of view, with comparable plate-fin surfaces in which the fins are of continuous or louvered-sheet form.

Nomenclature

The following nomenclature is used in the paper:

A = total heat-transfer area, including both extended and direct surface, sq ft
A_c = minimum free-flow area, sq ft
A_f = extended or fin (pin) area, sq ft
a = fin cross section, ft²
c_p = specific heat at constant pressure, Btu/(lb deg F)
D = pin diameter, ft
f = friction factor, defined by Equation [4], dimensionless
g_c = proportionality factor in Newton's second law, 32.2 (lb/#)(ft/sec²)
G = mass velocity, based on minimum free-flow area, lb/(hr sq ft)
h = unit conductance for convection heat transfer, Btu/(hr sq ft deg F)
k = unit thermal conductivity, Btu/(hr sq ft deg F/ft)
L = heat-exchanger flow length, ft
l = length of fin from root to center of flow passage, ft
m = $\sqrt{hp/ka}$, 1/ft
n = frequency of alternate vortex shedding from circular tube, 1/sec
P = pressure, #/sq ft
p = fin perimeter, ft
r_h = hydraulic radius, A_cL/A, ft
V = velocity, fps
X_l = ratio of longitudinal tube spacing to tube diameter, dimensionless
X_t = ratio of transverse tube spacing to tube diameter, dimensionless
β = ratio of heat-transfer area to volume between plates, sq ft/cu ft
Δ = as a prefix, denotes "final minus initial"
δ = fin thickness, ft
η = fin effectiveness, dimensionless
ρ = fluid density, lb/cu ft
μ = fluid viscosity, lb/(hr ft)
ψ = fin perimeter, ft
N_R = Reynolds number based on hydraulic diameter, $4r_hG/\mu$
N_{R_d} = Reynolds number based on pin or tube diameter, DG/μ
$N_{R\psi}$ = Reynolds number based on fin perimeter, $\psi G/\mu$
N_{St} = Stanton number, $h/(Gc_p)$
$N_{St}N_{Pr}^{2/3}$ = Colburn heat-transfer grouping
$V/(nD)$ = Strouhal number

Introduction and Objectives

It has long been recognized that the performance of plate-fin heat-transfer surfaces can be improved by interrupting the surface at intervals so as to maintain thin boundary layers. In 1942 Norris and Spofford (1)[2] demonstrated the high conductances attainable with gas flow over interrupted fin surfaces of various types, although at the same time showing that high conductance by this method is bought at the expense of high friction factors. Since that time a number of papers, e.g., (2, 3, 4), have appeared giving basic test data for particular examples of commercially available high-performance surfaces. It also has been shown (5) that it is possible to design lighter-weight and smaller-volume heat exchangers, for a particular heat transfer and pressure-drop performance, with interrupted fin surfaces than with continuous fins, despite the high friction factors characteristic of such surfaces.

Interest in high-performance surfaces has centered principally on short-width fins formed from thin sheet material, generally called "strip-fin" or "louvered-fin" surfaces. For a given fluid velocity the heat-transfer conductance on such surfaces can be increased almost indefinitely by decreasing the width of the fin in the flow direction. Although the friction factors are high relative to continuous fins, the friction is principally skin friction or boundary-layer shear, providing that the fin thickness is small relative to the fin width. The friction factors in such surfaces are in turn very much lower than in surfaces in which the fins are blunt bodies on which boundary-layer separation can occur, with attendant form drag arising from pressure differences on the upstream and downstream sides of the fins.

Strip fins, however, have a limitation that greatly restricts their possibilities for certain applications. The high surface-heat-transfer conductance, attainable by scaling the fin down to small width and thickness, may be largely nullified by temperature drop within the fin itself because of small conduction cross section. This can best be shown by examination of a dimensionless parameter, the fin effectiveness η expressing the ratio of the heat

[1] Assistant Professor of Mechanical Engineering, Stanford University. Assoc. Mem. ASME.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, September 24, 1953. Paper No. 53—A-211.

[2] Numbers in parentheses refer to Bibliography at end of paper.

actually transferred through the fin root to the heat that would be transferred were the fin material of infinite thermal conductivity. For fins of constant conduction cross section, from (6)

$$\eta = \frac{\tanh ml}{ml} \quad [1]$$

where

$$m = \sqrt{\frac{hp}{ka}} \quad [2]$$

For a strip-fin where the fin thickness δ is small relative to width of the fin in the direction of flow

$$m = \sqrt{\frac{2h}{k\delta}} \quad [3]$$

The effectiveness decreases with increasing m. Thus the combination of high h and small δ, as the fin is scaled to smaller dimensions, tends toward low effectiveness. Where the fins can be made from high-conductivity material, such as copper or aluminum, very effective strip-fin surfaces have been constructed, but for high-temperature applications such as a gas-turbine regenerator, where it is desired to use high-alloy steels ($k \approx 12$, as compared to $k = 100$ for aluminum and 220 for copper), the effectiveness can become very low and the advantage of high heat-transfer coefficient h is largely lost.

Fin effectiveness can be increased by increasing δ, or, in general, by employing fins with large conduction cross section relative to surface area (high a relative to p in Equation [2]). This, however, means abandoning the thin-fin, or strip-fin geometry in favor of blunt fin sections, and accepting the very high friction factors associated with boundary-layer separation that must occur unless fins can be constructed in perfect streamlined shapes.

The simplest of the blunt bodies that can be used for fins, and also the shape having the highest conduction cross section relative to surface area, is the circular cylinder, or pin fin. Plate-fin surfaces employing pin fins generally are constructed from small-diameter circular wire extending between the plates. The pins are arranged in some kind of regular pattern and the flow is similar to flow normal to conventional tube banks. The principal difference is that the system is on a much smaller scale, and consequently operation is generally at very much lower Reynolds numbers than with conventional tube banks. Very small-diameter pins can be employed, yielding extremely high conductances with reasonably high fin effectiveness.

The objectives of this paper are (*a*) to present some test data on pin-fin surfaces, part of which are new and part of which have been published previously; (*b*) to discuss some of the difficulties encountered with these surfaces, both in fabrication and in their heat-transfer and flow-friction performance; and (*c*) to illustrate the possibilities and limitations of such surfaces in the design of heat exchangers. Consideration will be restricted entirely to pin fins of circular section, and to heat-exchanger applications involving heat transfer to or from gases.

Nondimensional Variables Employed for Correlation

For all of the test data to be discussed a common system of nondimensional parameters will be employed and, where necessary, data have been recomputed on this basis. For a heat-transfer parameter, the Colburn grouping, $N_{St}N_{Pr}^{2/3}$ is employed throughout. The friction factor f is defined as

$$\frac{\Delta P}{\rho} = f\frac{A}{A_o}\frac{G^2}{2g_c\rho^2} \quad [4]$$

where A is the total heat-transfer (and friction) area, including both fins and wall, and A_o is the minimum free-flow area. The mass velocity G, employed in both the friction-factor definition and in the Stanton number, is likewise based on the minimum free-flow area. This definition of friction factor is identical to the Fanning friction factor commonly used for flow inside cylindrical tubes, since in that case

$$\frac{A_o}{A} = \frac{r_h}{L} \quad [5]$$

The basic data for each surface consists of $N_{St}N_{Pr}^{2/3}$ and f presented as functions of a Reynolds number. The Reynolds number without subscript, N_R, is defined as follows

$$N_R = \frac{4r_hG}{\mu}$$

where r_h is defined by Equation [5].

A Reynolds number based on pin diameter D also will be employed

$$N_{Rd} = \frac{DG}{\mu}$$

The pin-fin data of Norris (1) originally were presented employing a Reynolds number based on pin perimeter ψ

$$N_{R\psi} = \frac{\psi G}{\mu} = N_{Rd}\,\pi$$

Previous Work

The first basic heat-transfer and flow-friction data for pin-fin surfaces are those of Norris and Spofford (1). They used 0.04-in-diam pins arranged in an in-line pattern as shown in Fig. 1. They report that the ratio of fin area to wall area was 3.18, so that the plate spacing was about $^3/_4$ in. The fin effectiveness was taken into consideration employing Equation [1]. In evaluating the heat-transfer conductance and friction factors, Norris and Spofford report that they used only the surface area of the pins rather than the total area. Since the pins comprised 76 per cent of the total area, and the conductance on the pins can be expected to be very much higher than on the wall area, their data probably represent closely the behavior of the pins alone. For use in later comparisons a summary of their test data is given in Table 1.

TABLE 1 NORRIS AND SPOFFORD PIN FINS

$N_{R\psi}$	$N_{St}N_{Pr}^{2/3}$	f	N_{Rd}
300	0.052	0.184	95.5
500	0.043	0.174	159
1000	0.033	0.164	318
2000	0.025	0.152	637
4000	0.019	0.142	1274

The Norris and Spofford data compare quite favorably with most test data for flow normal to banks of in-line circular tubes, although the Reynolds-number test range is considerably lower.

The test performance is presented for two other pin-fin surfaces in reference (7). These surfaces are designated as AP-1 and AP-2. The pin-layout diagrams are shown in Fig. 1 and the basic test data are presented in Figs. 2 and 3. These surfaces were built by production methods; 0.04-in-diam wire was bent to form rows of pins which were then assembled between parallel plates and attached by furnace brazing. The Norris and Spofford pin-fin surface had been constructed from pins protruding through drilled plates, a method that may not be feasible for production work. Surface AP-1 had the same pin diameter and layout diagram as the Norris and Spofford surface, but the plates were spaced at only 0.24 in., with the result that the pins comprised only 51 per cent of the total heat-transfer area; i.e., about half

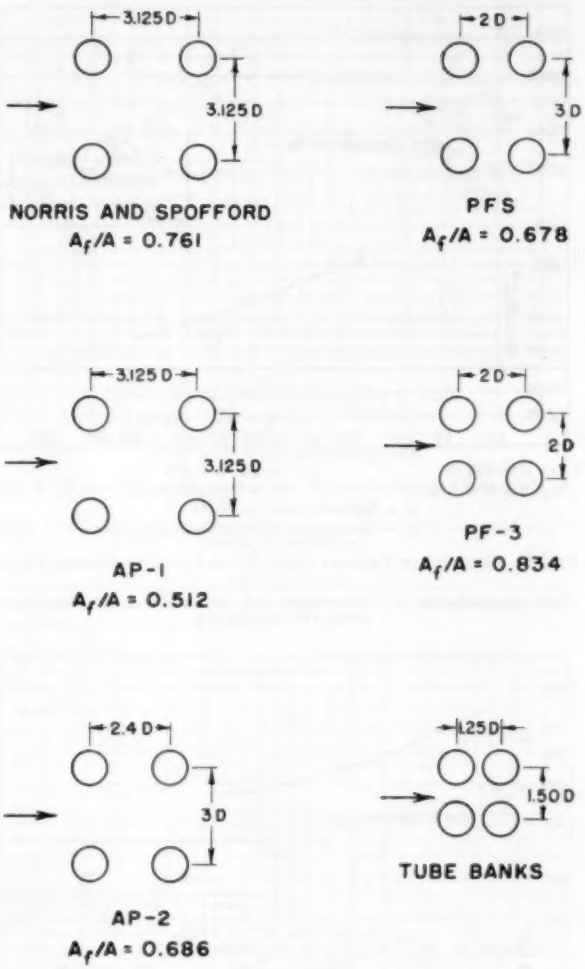
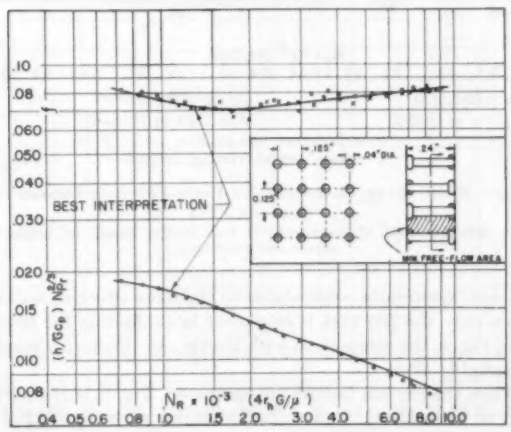


FIG. 1 PIN CONFIGURATIONS UNDER CONSIDERATION



r_h = 0.00361 ft A_f/A = 0.512
N_{R_d}/N_R = 0.231 β = 188 sq ft/cu ft
o = hot-core test points
x = isothermal friction factors

FIG. 2 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, SURFACE AP-1
(Best interpretation of heat-transfer test results based on estimate of steam-side resistance.)

of the total area was relatively ineffective plate or wall area. In surface AP-2, a closer pin spacing was employed, Fig. 1, in order to increase the pin-area ratio to about 69 per cent. In addition, corrugated material was employed for the plates in order to effect a better brazing bond between the bent-pin base and the plates, Fig. 3; difficulty was encountered with pins coming loose in AP-1.

These two surfaces were tested at the U. S. Naval Engineering Experiment Station, Annapolis, Md., in a test apparatus and by methods fully described elsewhere (2). A summary of the basic test data, from the full lines of Figs. 2 and 3, is given in Tables 2 and 3.

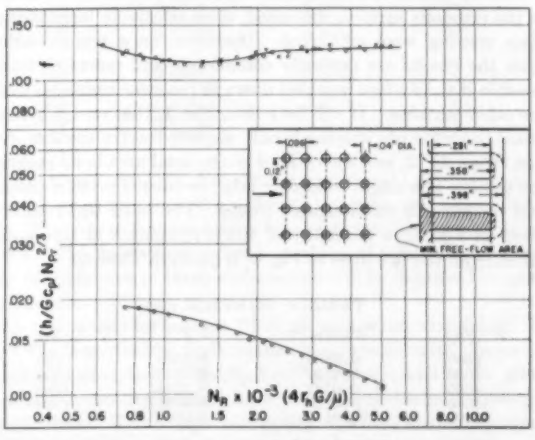
TABLE 2 SURFACE AP-1

N_R	$N_{St}N_{Pr}^{2/3}$	f	N_{R_d}
800	0.0183	0.0800	185
1000	0.0175	0.0755	231
1250	0.0163	0.0722	289
1500	0.0152	0.0708	346
2000	0.0134	0.0707	462
2500	0.0123	0.0725	578
3000	0.0116	0.0741	693
4000	0.0105	0.0762	924
5000	0.00970	0.0780	1156
6000	0.00905	0.0793	1387
8000	0.00808	0.0815	1850
9000	0.00768	0.0821	2080

TABLE 3 SURFACE AP-2

N_R	$N_{St}N_{Pr}^{2/3}$	f	N_{R_d}
600	....	0.131	170
800	0.0188	0.120	227
1000	0.0181	0.116	264
1200	0.0174	0.114	341
1500	0.0165	0.115	426
2000	0.0151	0.121	568
2500	0.0140	0.124	710
3000	0.0131	0.125	852
4000	0.0117	0.127	1136
5000	0.0108	0.127	1420

A comparison of $N_{St}N_{Pr}^{2/3}$ for the Norris and Spofford, AP-1, and AP-2 surfaces on a diameter Reynolds-number basis is shown in Fig. 7. The difference is principally due to the fact that the performance of the latter two surfaces was evaluated on the basis of the total heat-transfer area rather than the pin area only. The performance is quite close when AP-1 and AP-2 are recalculated on the basis of pin area alone, Fig. 8. Although the plates in



r_h = 0.00293 ft A_f/A = 0.686
N_{R_d}/N_R = 0.284 β = 204 sq ft/cu ft
o = hot-core test points
x = isothermal friction factors

FIG. 3 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, SURFACE AP-2
(Best interpretation of heat-transfer test results based on estimate of steam-side resistance.)

AP-2 are corrugated, they are evidently no more effective for heat transfer than smooth plates because of the stagnant-fluid region between the corrugations. A comparison of the friction factors in Fig. 9 shows similar behavior (total-area basis for AP-1, AP-2). However, if the AP-2 curve is corrected to pin area only (divide by A_f/A) it will lie above the Norris and Spofford curve, indicating that the corrugated-wall area does contribute considerably to the friction.

The difficulty with these early attempts at construction of pin-fin surfaces by production methods is that the area compactness of the pins (heat-transfer area per unit of volume) is so low that the advantage of high conductance per unit of area is largely lost. It can be argued easily that the rows of pins in AP-1 and AP-2 could be replaced by, say, 0.010-in. strip fins with interruptions every $^1/_4$ in., and the conductance on a volume basis would not be materially different from what is attained with the pin-fin surfaces. Since the form drag largely would be eliminated the performance on an over-all heat-transfer friction-power basis might be considerably better. This has been demonstrated.[3]

New Experimental Data

More recent work with pin-fin surfaces has been aimed toward closer pin spacing to give a higher area compactness, and elimination of the awkward pin-to-plate connection of surface AP-2 which apparently accounts for a substantial portion of the high friction factors of that surface. Two pin-fin surfaces, designated PFS and PF-3, have been tested as a part of the Office of Naval Research compact heat-exchanger program at Stanford University. Both were tested in a steam-to-air test duct (8). The pin patterns are shown in Fig. 1 and the surfaces are described in more detail in the diagrams in Figs. 4 and 5.

Surface PFS was built in the laboratory shop by hand methods. It was a single-tube test core built up of two parallel brass plates spaced $^7/_8$ in. apart, drilled to take $^1/_8$-in. brass rods in the pattern shown. The pins were soldered to the plates, and provision was made to measure the plate temperature directly, thus eliminating the necessity of estimating the steam-side resistance, as was the case for the AP-1 and AP-2 core tests. Pressures were measured at a number of points along the plate, eliminating the necessity of estimating the entrance and exit pressure losses.

This surface in itself is not a compact surface in the same sense as the previous surfaces discussed, since relatively large pins and plate spacing were employed. However, on a nondimensional basis the results are perfectly comparable and represent what is possible if the surface is scaled down to pin sizes comparable with the other surfaces. It will be noted, Fig. 1, that on a nondimensional basis the pin spacing is only slightly less (in the flow direction) than AP-2, and the ratio of pin to total area is substantially the same. The major difference is in the clean fin-root connection and the smooth uncorrugated plates. The basic test results are given in Fig. 4; a summary of the performance of this surface, taken from the full lines in Fig. 4, is given in Table 4.

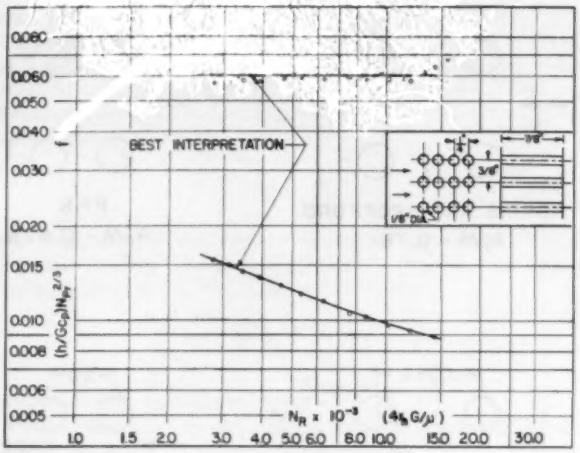
TABLE 4 SURFACE PFS

N_R	$N_{St}N_{Pr}^{2/3}$	f	N_{R_d}
2500	0.0163	0.0605	780
3000	0.0152	0.0605	936
4000	0.0137	0.0605	1248
5000	0.0126	0.0605	1560
6000	0.0117	0.0605	1873
8000	0.0105	0.0605	2495
10000	0.00972	0.0605	3120
12000	0.00920	0.0605	3745
15000	0.00875	0.0605	4680

The comparison of the heat-transfer performance of this surface with AP-2 in Fig. 7 indicates an improvement of about 20 per cent, apparently attributable to the more effective wall area. When only the pin area is employed in evaluating the conductance, Fig. 8, the comparison with Norris and Spofford's results is remarkably close.

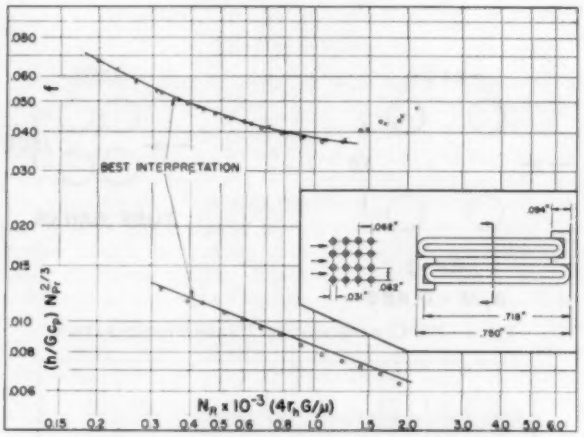
When the friction factors are compared, Fig. 9, surface PFS is seen to have about $^1/_2$ the friction factor of surface AP-2. In general, it is found that with flow normal to in-line tube banks the friction factor is lowered by decreasing the cylinder spacing in the longitudinal direction, since the form drag of a cylinder is decreased if it lies in close to the wake of a preceding cylinder. This unquestionably accounts for the low friction factors of the PFS surface as compared to the Norris and Spofford surface (to be accurately comparable in Fig. 9, the Norris and Spofford fric-

[3] Reference (7), p. 61.



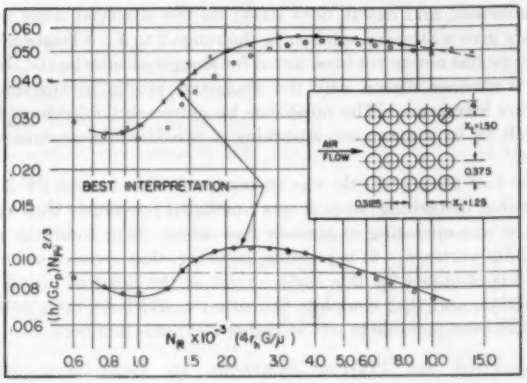
$r_h = 0.00835$ ft $\quad A_f/A = 0.678$
$N_{R_d}/N_R = 0.312 \quad \beta = 74$ sq ft/cu ft
o = hot-core test points
x = isothermal friction factors

FIG. 4 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, SURFACE PFS
(Best interpretation of heat-transfer test results based on estimate of steam-side resistance.)



$r_h = 0.00134$ ft $\quad A_f/A = 0.834$
$N_{R_d}/N_R = 0.482 \quad \beta = 339$ sq ft/cu ft
o = hot-core test points
x = isothermal friction factors

FIG. 5 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, SURFACE PF-3
(Best interpretation of heat-transfer test results based on estimate of steam-side resistance.)



$X_t = 1.50$; $X_l = 1.25$; $N_{Rd}/N_R = 1.256$
$D = 0.25$ in.; $r_h = 0.00415$ ft
o = hot-core test points
x = isothermal friction factors

FIG. 6 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, IN-LINE TUBE BANKS
(Best interpretation of heat-transfer test results based on estimate of steam-side resistance.)

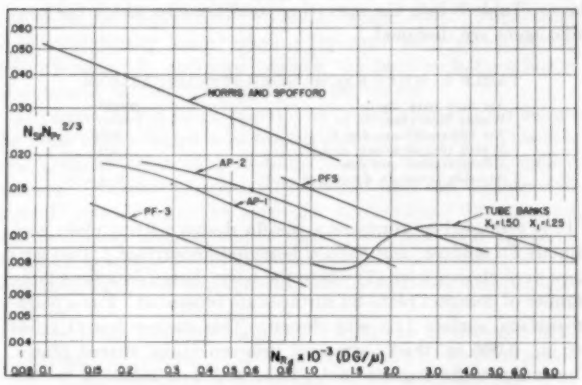


FIG. 7 COMPARISON OF HEAT-TRANSFER PERFORMANCE ON A DIAMETER-REYNOLDS NUMBER BASIS

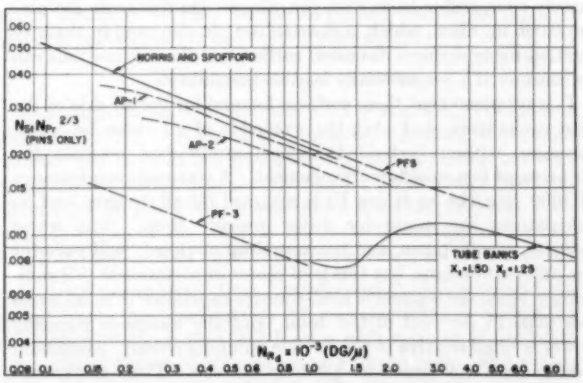


FIG. 8 COMPARISON OF HEAT-TRANSFER PERFORMANCE WITH CONDUCTANCE BASED ON PIN AREA ONLY

tion curve should be lowered by 24 per cent, since the friction factor was based on pin area rather than total area), but can hardly account for the big difference between PFS and AP-2. The major difference is apparently the price paid for the corrugated-wall surface, and indicates that some better production method must be found for effecting a bond between the pins and the plates before this kind of heat-transfer surface becomes commercially attractive.

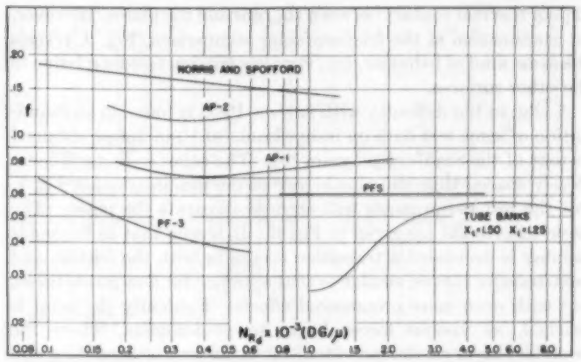


FIG. 9 COMPARISON OF FRICTION FACTORS ON A DIAMETER-REYNOLDS NUMBER BASIS
(Friction factor based on total area.)

Surface PF-3 is of brazed-aluminum construction and was manufactured by production methods. It is an extremely compact surface with close pin spacing, Fig. 1. The diagram in Fig. 5 gives the dimensions and details of construction. The pins are 0.031 in. diam, and small aluminum channels were used for pin spacing and to help effect a good thermal bond in the brazing operation. The channels were arranged parallel to the flow to avoid some of the difficulties of AP-2, and the ratio of pin-to-wall area, 0.834, is sufficiently high so that the behavior of the wall area is of little importance. The pins were formed from wire wound in flat coils.

The test core was of multiple-tube construction, similar to test cores described in reference (8). It was tested in a steam-to-air test duct by methods fully described in (8). The basic test results are given in Fig. 5 and are summarized in Table 5.

TABLE 5 SURFACE PF-3

N_R	$N_{St}N_{Pr}^{2/3}$	f	N_{R_d}
200	.....	0.0673	96.3
250	.....	0.0600	120
300	0.0132	0.0551	145
400	0.0118	0.0491	193
500	0.01085	0.0454	241
600	0.01020	0.0430	289
800	0.00912	0.0399	385
1000	0.00840	0.0383	482
1200	0.00784	0.0373	578
1500	0.00720	....	722
2000	0.00645	....	963

It will be noted in Fig. 5 that at about $N_R = 1400$ the friction factors start to rise abruptly. Similar behavior will be noted for PFS. The rise in friction factor was accompanied by an intense whistle. This behavior imposes something of a limitation on the range of usefulness of these surfaces and will be discussed in more detail later.

The test results for surface PF-3 are somewhat surprising. In Fig. 7 the heat-transfer performance is compared with the previous surfaces discussed. The $N_{St}N_{Pr}^{2/3}$ curve lies well below all of the others, indicating a much lower unit conductance than would be anticipated. The comparison in Fig. 8, where the conductance for all of the surfaces is evaluated on the basis of pin area only, is even more striking. All of the previous surfaces correlate quite closely and the maximum spread of 20 per cent can be attributed to effects of wall area and differences of pin pattern already discussed. Surface PF-3 yields conductances that are only about one third as high as the others. Since a few of the pins were loose, it was at first thought that this behavior might be the result

of poor thermal contact between the pins and the plates. However, an examination of the friction-factor comparison, Fig. 9, reveals the same kind of behavior, i.e., very low friction factors relative to the other surfaces.

A clue to the difficulty with surface PF-3 is found in an examination of some test data on in-line banks of 1/4-in. tubes, obtained as part of the Stanford program (9). The tubes were much more closely spaced than the pins in any of the pin-fin systems, Fig. 1, but the test performance will serve to illustrate the point. The basic test results are given in Fig. 6. It is seen that as Reynolds number is decreased a transition occurs in both the friction and heat-transfer curves, similar to that obtained for flow inside tubes, but with even more pronounced effects. Evidently the point is reached, as viscous forces begin to predominate, where the boundary layer on the individual tubes, instead of separating and shedding vortexes, carries over to the next tube and the entire flow becomes simple laminar motion. It is significant that the friction factor decreases more than the heat-transfer conductance, which would be expected as the separation and attendant form drag disappear.

The performance of the in-line tube banks has been plotted in Figs. 7, 8, and 9, and immediately suggests that surface PF-3 is operating in the laminar-flow region over its entire test range. When laminar flow of this sort occurs the advantage of pin fins is almost completely lost, since the performance is not appreciably better than can be attained with continuous fins.

The behavior of PF-3 indicates that there is a minimum practical limit to the pin pitch that may be used in pin-fin surfaces with in-line pin arrangements. Near this limit the performance is evidently quite sensitive to pin pitch, since surface PFS with a 2D × 3D arrangement showed no transition tendency in the Reynolds-number range tested, whereas surface PF-3 with a 2D × 2D arrangement was completely in laminar flow in the test range. It is true that surface PFS was tested at a higher range of Reynolds numbers than the others, and extrapolation of the tests of this surface to lower Reynolds numbers may be very risky. The decrease in slope of the AP-1 and AP-2 curves at low Reynolds numbers, Figs. 7 and 8, may indicate the start of transition.

If it is desired to have a very compact pin-fin surface, and operate at the low Reynolds numbers that are characteristic of applications of these surfaces, it appears that it would be better to employ a staggered pin arrangement rather than the in-line arrangements considered here. Test results for flow normal to staggered tube banks at low Reynolds numbers (9, 10) indicate that a transition of the type shown in Fig. 6 does not occur. There is a gradual transition as the separation point on the tubes moves toward the rear of the tubes, but a stagnant region between tubes apparently does not occur.

Pin Vibration

In the discussion of the performance of surface PF-3 an objectionable whistle, accompanied by an abrupt rise in friction factors, was mentioned. Similar behavior was noted for surface PFS at the upper end of the Reynolds-number test range. The in-line tube banks, Fig. 6, started to whistle at $N_R = 9000$, and it will be noted that again the friction factors increased, although not so abruptly. Measurements of the sound frequency indicate that it corresponds approximately to the frequency range of the early modes of transverse vibration of the pins or tubes. This aerodynamically induced vibration is believed to be due to a resonant condition developing between the frequency of the alternately shedding vortexes from the tubes and the natural bending frequency of the pins or tubes. For single tubes, the Strouhal number (V/nD) determines the frequency of vortex shedding (11). A Strouhal number of about 5 is noted for single tubes (11). These tests indicate that banks of in-line tubes exhibit a similar phenomenon, and calculations based on the observed noise frequency give a Strouhal number in the range 2 to 5. Vibration of this type has not as yet been noted for staggered tube banks, and this is another reason why the staggered pin-fin arrangement appears attractive. The noise can be extremely objectionable, as well as in some cases absorbing a considerable amount of power.

The fact that a whistle was encountered with surface PF-3 is somewhat disturbing, since it was concluded previously that this surface was operating in laminar flow where there would be no shedding vortexes. It is possible, however, that a resonant condition is obtained for this surface right at the point of incipient transition, and that complete transition would have been noted if it had been possible to test at higher Reynolds numbers.

Application of Pin-Fin Surfaces to Heat-Exchanger Design

In order to illustrate the potential advantages of the use of pin-fin surfaces in heat-exchanger design, as well as certain limitations, some comparative heat-exchanger designs have been made using a pin-fin surface and two other surfaces. For this comparison a simple single-pass cross flow air-to-water heat exchanger was chosen. The operating conditions specified correspond roughly to the specifications for an intercooler for a 500-hp gas-turbine plant. Table 6 lists the operating conditions to which the heat exchangers are designed.

TABLE 6 HEAT-EXCHANGER SPECIFICATIONS

Air flow rate, lb/hr	20000
Water flow rate, lb/hr	19860
Air temperature, deg F	200
Water temperature, deg F	60
Effectiveness, per cent	90
Air-side pressure drop, per cent	2.3

It is proposed to compare a pin-fin design with a plain continuous fin surface, and with a louvered-fin surface. Two surfaces were chosen from (7) where the basic design data for a large number of compact plate-fin surfaces are presented. For a plain-fin surface, surface 11.1 was chosen. This surface has 11.1 fins per in., 0.006 in. thick, arranged between 1/4-in. spaced plates. The louvered-fin surface is 3/8-11.1, which is identical to the plain-fin surface except that the fins are louvered at 3/8 in. intervals in the flow direction. For a pin-fin surface, PFS has been scaled down to a 1/8-in. plate spacing to put it on an approximately comparable basis with the others. On this basis the pins are 0.036 in. diam, which is close to the pin size used in the production-made surfaces discussed earlier, and this surface thus can be considered a commercially feasible possibility.

It is proposed that these surfaces be used for the air side of the heat exchangers, and that the waterside of all three be 1/8-in. passages, without extended surface, sandwiched between the air passages in a crossflow arrangement. A waterside conductance of 1000 Btu/(hr sq ft deg F) is assumed for all designs, and no calculations are made for water pressure drop. The water velocity, conductance, and pressure drop obviously will vary for the different designs, but this is of minor importance in this comparison since the waterside heat-transfer resistance is in all cases less than 15 per cent of the total, and the waterside pumping power is small relative to the air-side pumping power. Aluminum construction is assumed for all three surfaces. These surface arrangements are illustrated in Fig. 10 and the pertinent geometrical data are given in Table 7.

The designs were accomplished by a successive-approximation procedure similar to that described in reference (5). The results, assuming a square air-side frontal area, are shown in Fig. 10. Table 8 gives some of the pertinent information about these designs.

TABLE 7 SURFACE ARRANGEMENTS

	PFS	3/8 − 11.1	11.1
Hydraulic diameter, $4rh$, ft	0.01022	0.01012	0.01012
Fin area to total area, sq ft/cu ft	0.68	0.76	0.76
Free-flow to frontal area	0.420	0.587	0.587
Air-side heat-transfer area per unit of volume sq ft/cu ft	164.8	232	232
Waterside heat-transfer area per unit volume, sq ft/cu ft	60.8	60.8	60.8
Specific weight of core volume, lb/cu ft	23.5	15.93	15.93

TABLE 8 HEAT-EXCHANGER DESIGNS[a]

(Air side AP/P = 2.3 per cent, effectiveness = 90 per cent)

	PFS	3/8 − 11.1	11.1
Volume, cu ft	1.67	2.08	3.37
Weight, lb	39.2	33.2	53.7
Air-side frontal area, sq ft	3.24	2.22	1.89
Air flow length, ft	0.514	0.937	1.778
Air-side area, sq ft	275	483	782
Air-side conductance, Btu/(hr sq ft deg F)	70.2	37.6	22.0
Fin effectiveness	0.97	0.98	0.99
Air-side Reynolds number	3000	3100	3640
Air-side friction factor	0.0605	0.032	0.0107
Air-side average velocity, fps	64.0	66.8	78.1

[a] See Fig. 10.

The pin-fin surface, PFS, yields a design of smaller volume than either of the other two, although the weight is higher than the 3/8–11.1 design. It is interesting to note how the pin-fin design results in the same pressure drop as the others, although having a much higher friction factor. This is brought about by operating at a substantially lower velocity (pressure drop varies as the velocity squared), and also because the heat exchanger has a considerably shorter flow length than the others.

Although the pin-fin surface results in a smaller-volume design the price paid is larger core frontal area, see Fig. 10. This is a general characteristic of applications of compact high-performance surfaces. Whether or not a large frontal area can be tolerated depends entirely upon the particular application. Since frontal area is directly proportional to specific-flow rate, it is apparent that compact high-performance surfaces are more attractive for low-specific-flow-rate applications than for high-flow-rate applications.

These design comparisons indicate what can be attained with pin-fin surfaces providing that reasonably clean fin-to-plate bonds can be accomplished on a production basis. A surface like AP-2 with extremely high friction factors would yield a design of about the same size as the louvered-fin surface, but with a considerably larger frontal area. The designs do not show the pin-fin surface to be distinctly superior to the commonly employed types of plate-fin surface, but it is definitely competitive.

Summary and Conclusions

This paper includes a summary of the available test data on compact pin-fin heat-exchanger surfaces, including both heat-transfer and flow-friction performance. Consideration has been restricted to in-line pin arrangements. It is shown that the circular pin fin provides one method for obtaining very high heat-transfer conductances while at the same time maintaining high fin effectiveness. The more important conclusions are as follows:

1 Good performance from a pin-fin surface depends to a large extent upon the practicability of manufacturing a surface by production methods that will have a high ratio of pin-to-plate area, and at the same time effecting a pin-to-plate bond that is reasonably clean aerodynamically. The use of plate corrugations or channels transverse to the flow may result in prohibitively high friction factors without a compensating increase in plate conductance.

2 With in-line pin arrangement, a pin spacing closer than $3D$ in the transverse direction and $2D$ in the longitudinal direction may not be practicable because of the tendency of the flow to become completely laminar at the low Reynolds numbers where these surfaces probably will find their principal application.

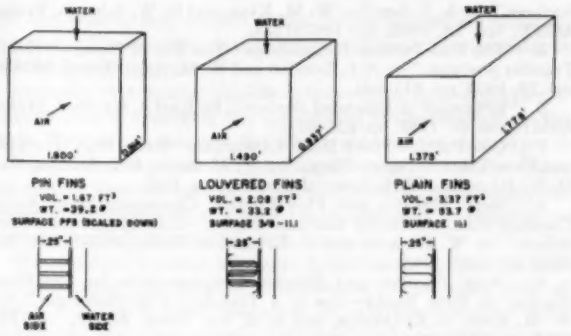


Fig. 10 Comparison of Heat Exchangers Designed for Identical Service From Various Types of Compact Surfaces

3 In-line pin-fin surfaces show a tendency toward pin vibration, which is believed to be a Strouhal-number effect. The critical Strouhal number appears to be in the range 2 to 5. This vibration results in an intense whistle as well as a noticeable increase in friction factor.

4 Heat-exchanger design studies indicate that despite the high friction factors characteristic of pin-fin surfaces, it is possible to design heat exchangers with pin-fin surfaces that are competitive with comparable plain-fin or louvered-fin surfaces, but that the resulting heat exchangers will have characteristically a larger frontal area.

It is quite possible that the laminar-flow and pin-vibration difficulties encountered with in-line pin arrangements could be eliminated or alleviated by the use of staggered arrangements, although this will probably also result in heat exchangers with still larger frontal areas because of the very high friction factors characteristic of staggered tube banks. In the long run it may prove worth while to investigate the possibility of employing pins with a streamlined cross section in order to attempt to eliminate a large part of the form drag of the circular section. Employing wire drawn or stamped to a streamlined section may not be out of the question.

Acknowledgments

The Office of Naval Research, the Bureau of Ships, and the Bureau of Aeronautics are currently sponsoring a research program on compact heat-transfer surfaces at Stanford University. The author prepared this paper as a part of this program and expresses his appreciation to the sponsoring organizations.

Appreciation is due also to the Air Preheater Corporation, and to the Harrison Radiation Division of the General Motors Corporation, who supplied test cores for this investigation.

Mr. George C. Wilson and Mr. Ivan Bjorklund, Stanford University graduate students, were of invaluable assistance in the test work, calculations, and preparation of illustrations.

Bibliography

1 "High-Performance Fins for Heat Transfer," by R. H. Norris and W. A. Spofford, Trans. ASME, vol. 64, 1942, pp. 489–496.

2 "Test Results of High-Performance Heat Transfer Surfaces Used in Aircraft Intercoolers, and Their Significance in Gas-Turbine Regenerator Design," by A. L. London and C. K. Ferguson, Trans. ASME, vol. 71, 1949, pp. 17–26.

3 "Heat-Transfer and Flow-Friction Characteristics of Some Compact Heat-Exchanger Surfaces," by A. L. London and W. M. Kays, Trans. ASME, vol. 72, 1950, pp. 1087–1097.

4 "Heat-Transfer and Flow-Friction Characteristics of Some Compact Heat-Exchanger Surfaces, Part 3—Design Data for Five

Surfaces," by A. L. London, W. M. Kays, and D. W. Johnson, Trans. ASME, vol. 74, 1952, pp. 1167–1178.

5 "The Gas-Turbine Regenerator—The Use of Compact Heat-Transfer Surfaces," by A. L. London and W. M. Kays, Trans. ASME, vol. 72, 1950, pp. 611–621.

6 "Efficiency of Extended Surface," by Karl A. Gardner, Trans. ASME, vol. 67, 1945, pp. 621–631.

7 "Gas-Turbine-Plant Heat Exchangers—Basic Heat Transfer and Flow-Friction Design Data," by W. M. Kays, A. L. London, and D. W. Johnson, ASME Research Publication, 1951.

8 "Heat Transfer and Flow-Friction Characteristics of Some Compact Heat-Exchanger Surfaces Part 1—Test System and Procedure," by W. M. Kays and A. L. London, Trans. ASME, vol. 72, 1950, pp. 1075–1085.

9 "Heat Transfer and Friction Characteristics for Gas Flow Normal to Tube Banks—Use of a Transient-Test Technique," by W. M. Kays, A. L. London, and R. K. Lo, Trans. ASME, vol. 76, 1954, pp. 387–396.

10 "Heat Transfer and Fluid Friction During Flow Across Banks of Tubes—IV" (A Study of the Transition Zone Between Viscous and Turbulent Flow), by O. G. Bergelin, G. A. Brown, S. C. Doberstein, Trans. ASME, vol. 74, 1952, pp. 953–960.

11 "Dynamics of Real Fluids," by E. G. Richardson, Edward Arnold and Company, London, England, 1950.

Discussion

DAVID ARONSON.[4] The author has added another stone to the monumental structure proclaiming the merits of compact heat exchangers. Yet, despite the impressiveness of his work and that of his associates, as well as the earlier studies of Norris and Spofford,[5] only a limited number of compact heat exchangers have been built for gas-turbine service, either in this country or abroad. The original paper of Norris and Spofford is over 10 years old. Although their work was supported by the General Electric Company, that company, for its industrial gas-turbine applications, has chosen to use bulky shell-and-tube design rather than the type of compact surfaces reported on in this and previous papers.

Many explanations for the slow acceptance of this type of surface for gas-turbine application can be offered. The discussion will be limited to those features covered in the present paper. The author has clearly shown two pitfalls to be avoided in the design of pin-fin heat exchangers. He advises that:

1 Range of operation for the particular pin diameter and length should be such that vibration of the pins is not induced by the fluid flow.

2 Pins should be spaced in such a staggered pattern that the boundary layer does not carry over from pin to pin to give laminar-flow conditions.

Despite such foreknowledge, the regenerator manufacturer may run into these and other difficulties because the unit must operate over a wide range of flows, temperatures, and pressures, and because, in the actual construction, certain arrangements are dictated by the fabricating techniques at his command.

The paper is written with the view that there is a design of compact heat-transfer surface which represents the ideal. The quest for this ideal has provided us with the much-needed performance data on actual cores, for which we are all grateful. However, some of the comments accompanying the presentation of these data represent wishful thinking applied to one design or another which the author happens to favor. The present paper does not emphasize the gap which necessarily exists between the performance of hand-made models and commercial cores. In addition, the choice of core pattern for a particular gas-turbine application depends on so many more factors than can be covered in the paper that the offering of standards of performance as superior or inferior is likely to be misleading.

[4] Worthington Corporation, Harrison, N. J. Mem. ASME.

[5] Author's reference (1).

The present paper seems to indicate that the pin fin is to be preferred to the strip fin. In order to show a favorable picture for the pin fin the author compares a handmade, scaled-up model (three times normal size) of pin-fin design with a commercial core of louvered-fin design, as though the distinctions shown are related solely to geometric configuration. Irregularities associated with commercial production tend to lower performance. Moreover, there is no particular reason for assuming that the pattern chosen for the louvered-fin design represents an ideal arrangement. A fin spacing of 11.1 fins per in. is not the closest spacing which can be obtained. If a more compact unit were desired, a spacing of, say, 18 fins per in. could be obtained, and this would result in a core having a smaller volume and less weight than the pin-fin unit based on ideal construction.

The requirements of the regenerator installation, however, may call for a close or a wide fin spacing. In the case of pin-fin design, it appears that performance is extremely sensitive to pin spacing so that adjustment for the particular installation is not effected readily. In terms of actual manufacturing methods, the spacing of strip fin can be changed at negligible expense, whereas a change in pin spacing calls for a change in the basic dies.

The tests reported all show that a fairly wide spacing of the pins is desirable. The pin-fin design tested by Norris and Spofford used a spacing of over 3 diameters. The present paper indicates that a spacing of at least 2 diameters constitutes a minimum working figure. This dictum results in a surface having an appreciable proportion of area contributed by the separating plates.

With regard to the argument that pin fins give higher fin effectiveness than comparable strip fin, the writer feels that this claim does not stand up on analysis. If the distance between separating plates is considered fixed, then a certain cross section of metal is required to transfer heat at a particular rate with a set temperature gradient. The geometrical pattern of the fin does not enter into the analysis. An illustration will show how fin effectiveness depends on cross section of metal, and at the same time will show that the thinner the pin or the strip is, the better will be the overall performance for a given cross section. Suppose we consider two designs of the same geometrical pattern. Design (a) is made with pins 0.020 in. diam and design (b) with pins 0.010 in. diam. For the same total metal cross section, design (b) will have 4 times the number of pins as design (a), and twice the surface area. If the same heat rate for a given temperature difference is to be maintained in the two designs, then design (b) calls for one half the heat-transfer coefficient. The fin effectiveness then remains unchanged. The relationships are

$$\begin{aligned} Q_1 &= Q_2 \\ h_1 &= 2h_2 \\ \Delta T_1 &= \Delta T_2 \\ p_1 &= 2p_2 \\ a_1 &= 4a_2 \\ k_1 &= k_2 \\ l_1 &= l_2 \end{aligned}$$

The ratio between η_1 and η_2 (fin effectiveness) will be unity since m_1 equals m_2 and l_1 equals l_2

$$\left(\frac{\eta_1}{\eta_2}\right) = \left[\frac{(\tanh m_1 l_1)/(m_1 l_1)}{(\tanh m_2 l_2)/(m_2 l_2)}\right]$$

$$\left(\frac{m_1}{m_2}\right) = \left[\frac{(h_1 p_1)/(k_1 a_1)}{(h_2 p_2)/(k_2 a_2)}\right] = \left[\frac{(2h_2)(2p_2)/(k_2)(4a_2)}{(h_2 p_2)/(k_2 a_2)}\right] = 1$$

so that $\eta_1 = \eta_2$.

The lower heat-transfer coefficient called for in design (b) is obtained by reducing the mass velocity to about one third that of

design (*a*). The really significant gain secured by changing from design (*a*) to design (*b*) is that for the same weight of heat-transfer core the pressure drop has been reduced to about one fourth that of design (*a*). The price to be paid is the increased cost of building a core with the smaller-diameter pin, (even though the metal weight is unchanged). Further, the geometry of the final regenerator may be less favorable, in that the frontal area of design (*b*) will be 3 times that of design (*a*). The core volume will be the same for both designs.

Similar reasoning could be applied to the design of a strip or louvered-fin heat exchanger having the same total metal cross section. If a fin thickness of 0.010 in. is chosen, then the design would result in the same fin effectiveness as the 0.010-in-diam pins, provided the heat-transfer coefficient is adjusted to be the same as with the pins. The pressure drop with the strip-fin design might be higher than for the 0.010-in-diam pins but would be less than for the 0.020-in-diam pins. Diameter (or thickness) of section is thus seen to be more important than the shape of the section.

In conclusion, the writer wishes to express his appreciation of the fine job done by the author and his associates in building the scale model, testing the several cores, and preparing the performance charts. The author is doing this work in the hope that manufacturers will be encouraged to apply the concept of extended surface to the design of heat exchangers for gas-turbine service. This discussion has been written to assist in the selection of the appropriate core—the ideas expressed being the result of attempts to apply the data to regenerator designs. Of course, until actual units have been built, tested, and the results reported, these discussions will remain pretty much up in the air.

M. L. Ghai.[6] The paper gives an excellent presentation of pin-fin surfaces and their comparison to other types of surfaces used in heat exchangers of high performance.

Pin fins of small-diameter pins operating at low Reynolds number give high heat-transfer coefficients, but their large form drag gives abnormal pressure-drop characteristics, and they tend to have low compactness of the heat area per unit volume of the exchanger. The author's comparison of different surfaces on equal pressure-drop basis shows how the pin-fin exchanger may be designed better than an exchanger using plain fins or louver fins. However, it is to be noticed that the weight of the pin-fin exchanger is 39.2 lb, whereas that of an exchanger with louvered fins is 33.2 lb, indicating a larger material consumption for the pin-fin exchanger.

Have any experiments been carried out with pins having a large cross-sectional area at the root where they are connected to the plates and a comparatively smaller cross-sectional area in the middle; for instance, fins of conical pins instead of cylindrical pins? It is to be realized that the pin is heavily loaded near the root; i.e., the concentration of the heat flow per unit cross-sectional area is high; the loading near the middle is low. As a result, pins which provide the material where it is needed most require less material.

Referring to the aluminum pin fin of 0.036 in. diam and 0.25 in. length, mentioned by the author, the material consumption may be reduced by as much as 14 per cent by the use of a conical fin of same diameter at the root but negligible diameter in the middle. Comparison has been made with conical form as it is easier to calculate its performance. The use of the conical fin reduces the area by 50 per cent and the effectiveness is improved by about 1 per cent, so that the net heat-transfer area is reduced by about 49.5 per cent; but the material requirement and the volume and weight of the fin are reduced by about 66 per cent. The weight of the pin fin per unit heat transfer is reduced in this way.

Normally, it should be possible to prevent the reduction in heat-transfer area by enlarging the diameter of the conical pin at the root. However, in this case the effectiveness of the conical pin is so high (about 98 per cent) that the additional material increases the surface area without being accompanied by appreciable increase of effectiveness, and as a result, the cost of material to increase the heat transfer becomes too great.

The situation is somewhat more favorable in the case of steel fins operating with smaller effectiveness. The steel pins of 0.0625 in. diam in plates spaced $^1/_2$ in., may be compared with two conical steel pins—a conical pin having the same diameter at the root, and a conical fin having the same material requirement (Fig. 11 of this discussion). The conical fins give higher effectiveness. Heat transferred per fin is lower for conical fins since the heat-transfer area is less, except when the fins are working at comparatively low effectiveness, in which case the increased effectiveness due to the use of the conical shape overcomes the disadvantage of decreased area. The conical fins require less material for the same amount of heat transfer.

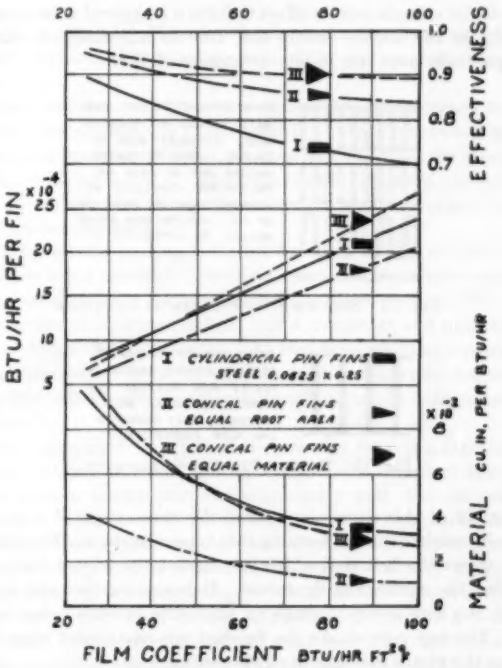

EFFECTIVENESS
BTU/HR PER FIN
MATERIAL CU. IN. PER BTU/HR
FILM COEFFICIENT BTU/HR FT²°F
I CYLINDRICAL PIN FINS STEEL 0.0625 x 0.25
II CONICAL PIN FINS EQUAL ROOT AREA
III CONICAL PIN FINS EQUAL MATERIAL

Fig. 11 Comparison of Typical Cylindrical Pin Fins With Conical Pin Fins of Varying Cross Section

The pin fins of varying cross-sectional area are likely to offer other advantages, such as, less pressure drop. The pin vibration and whistle, which offer serious limitation to the use of pin fins, also might be eliminated by pins of larger cross-sectional area at the root.

The mass production of pin fins of varying cross-sectional area is more difficult than the production of cylindrical pin fins, but not much more than the contemplated streamlined pin fins. The streamlining of circular wire by stamping or coining operations also can give fins of varying cross-sectional area—the streamlining may be done both in the direction of the air flow and in the direction normal to the air flow.

[6] Director, Research and Development, Bush Manufacturing Company, West Hartford, Conn.

SVEN HOLM.[7] The author is to be commended on the thorough work of investigating heat-transfer performance of pin-fin heat-exchanger surfaces. The results should be helpful in the design of extended-surface heat exchangers. The paper also stimulates thinking toward further developments and improvements of this type of heating surface.

As pointed out by the author, a pin-fin surface should have reasonably clean fin roots for high performance. The ideal pin-fin surface probably would be one where cylindrical pins project through the plates forming the flow passages and extend continuously through the exchanger core. This type of construction, however, does not lend itself very well to production methods. Further, if the heat exchanger is to function as a regenerator in a gas-turbine cycle, a different type of surface is desirable on the high-pressure air side.

Forming a continuous row of pins from wire stock appears to be the best solution from a production standpoint. With a thin ligament connecting the bases of the pins, the performance characteristics should be close to that of the ideal pin-fin surface. If the connecting ligaments between the pins are sufficiently thin, the pin-fin strands can be offset to form a staggered arrangement, which, as the author points out, has an advantage in surface compactness over the in-line arrangement.

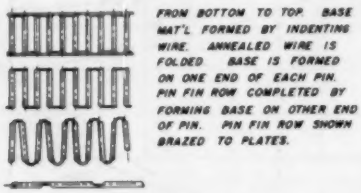

FROM BOTTOM TO TOP. BASE MAT'L FORMED BY INDENTING WIRE. ANNEALED WIRE IS FOLDED. BASE IS FORMED ON ONE END OF EACH PIN. PIN FIN ROW COMPLETED BY FORMING BASE ON OTHER END OF PIN. PIN FIN ROW SHOWN BRAZED TO PLATES.

FIG. 12 SEQUENCE OF PIN-FIN FABRICATION

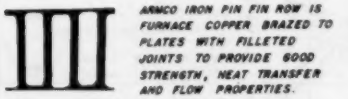

ARMCO IRON PIN FIN ROW IS FURNACE COPPER BRAZED TO PLATES WITH FILLETED JOINTS TO PROVIDE GOOD STRENGTH, HEAT TRANSFER AND FLOW PROPERTIES.

FIG. 13 BRAZED PIN-FIN SILHOUETTE

Fig. 12 of this discussion shows the three steps of a method actually worked out for forming this type of surface. The bottom view shows the first step which is to flatten the wire at the points forming the connecting ligaments. Between the first and second steps, the wire is run through an annealing furnace before bending. The top view shows the finished pin-row strand brazed between the plates forming the direct surface.

It is important that no reduction of pin area occur. Fig. 13 shows an enlarged view taken from a shadowgraph of the finished pin-row strand brazed between plates. The brazing material forms fillets at the pin base so that there is an increase in area at this point.

At a high rate of production of heat exchangers, the material weight and cost become an important item. It is therefore necessary to standardize on a plate spacing or fin length that gives the maximum heat transfer with a minimum weight of construction material over a range of mass velocities expected to be used. Fig. 14 shows a diagram to determine the optimum plate spacing for $^1/_{16}$-in-diam pins. In the range of mass velocities of gas in a gas-turbine regenerator, say, between 1.0 and 10 psf/sec, it can be seen that a plate spacing of $^1/_2$ in. is very satisfactory. This diagram is based on in-line arrangement of the pins. For a staggered arrangement and a given mass velocity, obviously the optimum would be a smaller plate spacing. However, with a given pressure drop the mass velocity over a staggered bank of pins would have to be lower and, therefore, the $^1/_2$-in. plate spacing is also satisfactory for this arrangement as well. As can be expected, the stainless-steel construction indicates smaller optimum plate spacing than that for carbon steel. However, in a gas-turbine-regenerator application, even with gas temperatures of 1100 to 1200 F, carbon steel might be used by treating the hot end of the heat exchanger in a hot aluminum dip.

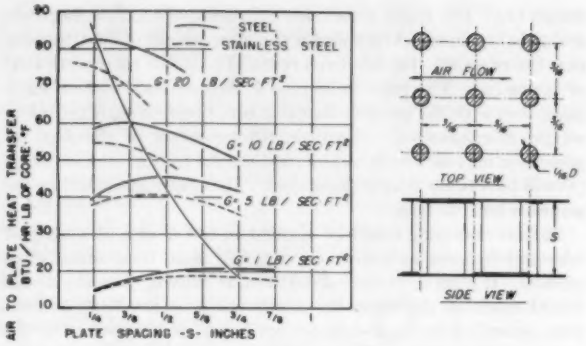


FIG. 14 OPTIMUM PLATE SPACING

The heating-surface comparison in Fig. 10 of the paper is interesting. The comparison is based on aluminum construction. However, if temperature conditions are such that the comparison be based on steel construction, the difference in size and weight between the pin-fin core and the $^3/_8$–11.1 core would be still greater as the fin effectiveness of the pin fins would be about 8 per cent higher than that for the louvered fins. With the aluminum construction, the louvered fins have about 1 per cent higher fin effectiveness. This, of course, is based on air-side conductances as given in Table 8 of the paper.

As mentioned by the author, a streamlined pin fin would be the answer to the problem of low area compactness, which is the disadvantage with cylindrical pin surfaces. By forming streamlined pins with an aspect ratio of 3 from the pin sections shown in Figs. 12 and 13, the circumference of the pin is increased about 28 per cent. The form drag is reduced very likely 50 to 60 per cent which makes it possible to use higher mass velocities for a given pressure drop and higher heat-transfer coefficients. Further, with a given free area to frontal area ratio, the number of pins per unit plate area will be increased.

Fig. 15, herewith, shows a comparison of fin effectiveness between circular and streamlined-shaped pins. It will be noted that for a film coefficient of 50 Btu/ft²-hr-deg F, the fin effectiveness of the streamlined pin is only 5 per cent lower than that for circular pin. With a circumference of 28 per cent higher for the streamlined pin, there is still an equivalent increase in heating surface

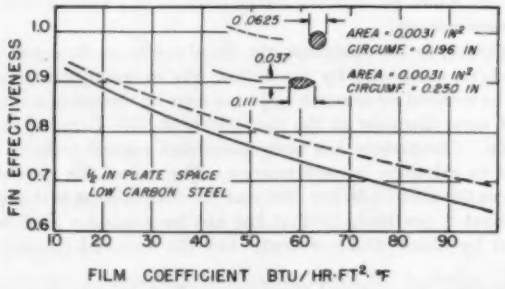


FIG. 15 EFFECT OF PIN SHAPE ON PIN EFFECTIVENESS

[7] Development Engineer, Heat Exchanger Division, Ferrotherm Company, Cleveland, Ohio. Mem. ASME.

of 23 per cent. This is based on steel pins and a plate spacing of $^1/_2$ in. With these improvements in view, it is reasonable to predict that a streamlined-pin core compared with the ones shown in Fig. 10 of the paper would have a frontal area of approximately that of the $^3/_8$-11.1 core. The flow length would be slightly more than that of the pin-fin core, with volume and weight considerably less.

In order to reduce the form drag to an absolute minimum, it is important that the trailing edge of the pin be sharp. If wire were drawn to a streamline shape, it is doubtful that a sharp trailing edge could be made. Coining the wire to a streamline shape after forming the pin-row strand seems to be the only way to obtain a sharp trailing edge.

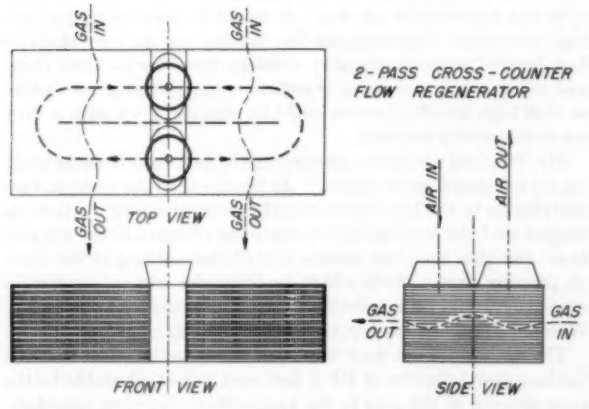
It is quite likely that the streamlined pin heating surface will solve the problem of pin vibration.

Fig. 16 of this discussion shows a set of shadowgraphs of streamline pin sections formed from $^1/_{16}$-in-diam wire with aspect ratios of approximately 3. The sections show various degrees of streamlining with section 7 being the most perfect. It is probably immaterial whether the entering edge is rounded or slightly pointed. In that case, section 3 would be the best choice, as it is easier to form than section 7. These sections are now being tested at Stanford University and both in-line and staggered arrangements of the pins will be investigated.

Fig. 17 shows a diagram of a regenerator with two cores in parallel. Each core has one pass on the gas side and two passes on the air side. The space between the two cores is the manifold for distributing the compressed air to the continuous strip-fin air passes. Up to an effectiveness of 75 per cent, this flow arrangement results in a well-proportioned design when using circular cross-section pin fins on the gas side. At the bottom of the illustration is shown a section through part of the core. A number of core blocks may be connected in parallel and connected to a common air header for any particular requirements of weight flow and effectiveness.

NO.	SHAPE	CIRCUM. Inches	WIDTH Inches	THICKNES Inches	ASPECT RATIO
1		0.253	0.1225	0.038	3.2
2		0.245	0.114	0.0365	3.1
3		0.250	0.1175	0.037	3.2
4		0.235	0.110	0.0365	3.0
5		0.228	0.1005	0.040	2.5
6		0.252	0.114	0.038	3.0
7		0.247	0.112	0.037	3.0

FIG. 16 STREAMLINED PIN SHAPES



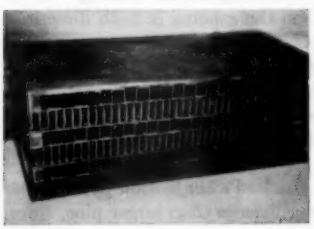


FIG. 17 PIN-FIN SURFACE-REGENERATOR CORE DESIGN

Will it be possible for the author to run some tests at higher Reynolds numbers to determine the range through which pin vibration occurs? It also would be interesting to learn how various pin and plate spacings affect the pin vibration and the friction factor. It has been the writer's experience that the friction factor may increase as much as 100 per cent in some cases in the critical range.

An investigation of the effect of plate spacing on heat transfer would be a very good addition to the excellent work already done by the author and his associates.

T. L. WOOLARD.[8] We have been conducting tests on pin-fin heating surfaces that parallel some of the work reported in this paper.

These tests have shown us two things that are contrary to the author's contentions: (1) That there is a gain to be made in using close plate spacing with the pin fins and (2) that the designer would be taking a risk in using data based on tests of $^1/_8$-in-diam pins for calculating the performance of heat exchangers using smaller-diameter pins.

We have been making tests on cores similar to the author's PFS core using nominal $^1/_8$-in-diam pins. In these test cores we have varied both the pin spacing and the plate spacing. We have used square pin spacings of 2, 3, and 4 diameters and plate spacings of 4, 8, and 12 diameters. At this time we have completed the pressure-drop tests for the full series and have heat-transfer tests for the 2-diameter pin spacing and part of the 3-diameter pin spacing.

These tests have shown that the pressure drop is a function of the pin spacing and that it is independent of the plate spacing. Since friction horsepower varies inversely with the amount of surface, it will be lower for closer plate spacing. Now, to make the wider plate spacing the more efficient combination, the heat transfer with closely spaced plates would have to be very much less than that for the wider spacing, but the tests have not shown this. Conversely, they have shown that the j-factor for a 4-diameter plate spacing is only 10 per cent less than that for a 12-diameter plate spacing, whereas there is a 29 per cent difference in surface for these plate spacings in combination with a 2-diameter pin spacing.

Thus for the range of spacings we have tested, the closer plate spacing will show a higher film coefficient - friction horsepower relationship than the wider plate spacing.

The writer does not agree with the author's conclusion relative to the cause of the low performance of the PF-3 test core, which is the close spacing of the pins in the longitudinal direction. This conclusion is not borne out by his PFS test core which has the same longitudinal spacing or by our tests in which we have had the same spacing in both directions. However, the surface is operating in the laminar-flow range but for what reason the writer does not care to hazard a guess. We have noticed in our own tests of 0.040-in-diam pins a tendency toward a transition at a Reynolds number based on pin diameter of approximately 400 which falls at about the mid-point in the PF-3 tests.

[8] Assistant to Chief Engineer, Air Preheater Corporation, Wellsville, N. Y.

We also have noticed that the smaller pins tend to have lower heat-transfer coefficients than the larger pins. For this reason and because of the chance of operating in a laminar-flow zone, it may be said that the designer would be taking a chance in designing with large-pin data when calculating small-pin performance, at least until the data are more complete.

We have also observed the pin-vibration phenomena with the $^1/_8$-in-diam pins. This has occurred at Reynolds numbers of 4000 for a 2-diameter-pin spacing, 5700 for 3-diameter-pin spacing, and 6300 for 4-diameter-pin spacing. These compare with the 4200 for the PFS test core. The foregoing Reynolds numbers are based on pin diameter.

Normally, a designer should not be troubled with this vibration as our experience has indicated that most applications will operate considerably below this point. However, our tests have shown that this pin vibration does not persist as flow is increased. We have found that, for the 3 and 4-diameter-pin spacings, the vibration stopped at Reynolds numbers between 9000 and 11,000. The effect persists over a greater range for the 2-diameter-pin spacing as our tests showed it at Reynolds numbers of 12,000 which was as high as we could operate.

We also have encountered pin vibration with smaller-diameter pins at correspondingly lower Reynolds numbers.

The author is to be congratulated for this paper and for his work in the other reports of the Stanford series which have contributed so much worth-while information to the field of gas-to-gas heat transfer. It is the writer's belief that the designer must have at his disposal a varied selection of surfaces from which to choose if he is to meet all of the conditions which are thrust upon him. To this end the author and his colleagues have made a significant contribution.

Author's Closure

The author wishes first to express his gratification for the large amount of discussion that this paper has aroused, and to thank the discussers for the time and effort that obviously they have all devoted to enhance the value of the paper. Each of the discussions will be treated in turn.

Mr. Aronson, unfortunately, seems to have derived certain impressions from the paper which certainly were not intended, but it is perhaps the fault of the author in that the objectives were not stated as clearly as they might have been. There is no intention to "sell" a particular heat-transfer surface, or to proclaim any particular surface as an ultimate ideal. The point of view which led to the preparation of the paper was roughly as follows: (*a*) It is well known that the pin-fin configuration provides a method of attaining extremely high heat-transfer conductances; (*b*) pin-fin surfaces are being used to a limited extent in industrial applications and are thus of technical interest; and (*c*) there are few basic heat-transfer and flow-friction data in the literature on this type of surface. Hence an analysis of the available data on pin-fin surfaces, and the presentation of new data appeared to be a worth-while project. There is no reason why this cannot be done in a perfectly objective manner, without bringing in all of the many considerations involved in the ultimate optimum design of, for example, a gas-turbine regenerator. The only purpose of presenting the heat-exchanger designs shown in Fig. 10 was to illustrate graphically the fact that the extremely high friction factors characteristic of pin-fin surfaces may not be a disadvantage in heat-exchanger design, providing that high friction factors are accompanied by high heat-transfer conductances. This is a feature of heat-transfer surfaces that often is not well understood. Obviously, there is no ultimate ideal of heat-transfer surface; the designer must make his choice for each application, based on literally dozens of independent considerations. In order to do so intelligently he must know something of the characteristic features of many types of surfaces, and above all, he must have some reliable basic heat-transfer and flow-friction data.

Mr. Aronson's discussion of fin effectiveness seems to the author to be an unnecessary complication of a basically simple problem. Obviously, this kind of argument can go on ad infinitum depending upon how one defines a "comparable strip fin." The author has chosen to define a comparable strip fin as one which has the same heat-transfer conductance, for a given flow velocity, as the pin fin. This means a strip fin whose length in the direction of flow is somewhat less than the pin-fin diameter. If one then defines a strip fin as a fin with a thickness that is small relative to flow length, it is quite apparent (see Equation [3]) that the pin-fin effectiveness always will be greater than that for the comparable strip fin.

For example, if the pin fins of Figs. 2 and 3 are replaced by comparable strip fins, these strip fins would only be a few thousandths of an inch thick, and in many applications would have a low fin effectiveness that completely nullifies the high heat-transfer conductance of such a fin. If the effectiveness is low the designer can elect to increase the fin thickness, but when the thickness becomes large relative to the flow length the fin becomes a blunt body with large form drag, and the author chooses not to call this a strip fin. The subject of this paper is one type of blunt-body fin. If fin effectiveness were never of any concern fins obviously should be made of the thinnest possible material in order to decrease form drag.

In answer to Mr. Ghai's question regarding pins of varying cross section, the author is aware of no experimental data on such surfaces, but they certainly offer an interesting possibility.

Mr. Sven Holm has presented a valuable addition to the paper in his discussion of fabrication methods that can and are being used to produce both circular and streamlined pin-fin surfaces. It is to be hoped that we will soon have some good basic heat-transfer and flow-friction data on such surfaces. The surfaces that Mr. Holm describes would appear to indicate that some of the earlier fabrication difficulties of pin-fin surfaces have been overcome. Of special interest are the streamlined fins and the indication that such surfaces may be feasible. The streamlined fin would combine some of the most attractive features of both strip fins and circular pin fins. It would be possible to get very high convection conductances (by making the fin very short in flow length) without excessive pressure drop due to form drag, and at the same time there is sufficient conduction cross section so that high fin effectiveness could be attained even with a very low-conductivity material.

Mr. Woolard's remarks concerning the influence of plate spacing do not stand up to reason. He implies that the plate surface contributes to the total convective heat-transfer rate—in fact, he implies that the plate surface is almost as effective as the pin surface—but that the plate surface contributes nothing to the overall pressure drop. Such a high-heat-transfer frictionless surface obviously would be a desirable thing and would lead one to Mr. Woolard's conclusions, but the author remains skeptical.

The author did not draw any conclusions to the effect that the laminar-flow behavior of PF-3 test core was attributable to the close spacing of the pins in the longitudinal direction, especially since the PFS test core had the same longitudinal spacing. What the tests do indicate is that when the spacing in both directions gets near $2D$ there is a danger of laminar-flow performance. This is an important point because one of the drawbacks of most of the pin-fin surfaces that have been built is poor area compactness as compared to comparable strip-fin surfaces. It is apparent that this deficiency cannot be overcome merely by cramming in more pins.

It is difficult to believe Mr. Woolard's contention that smaller pins have lower heat-transfer coefficients than larger pins, since,

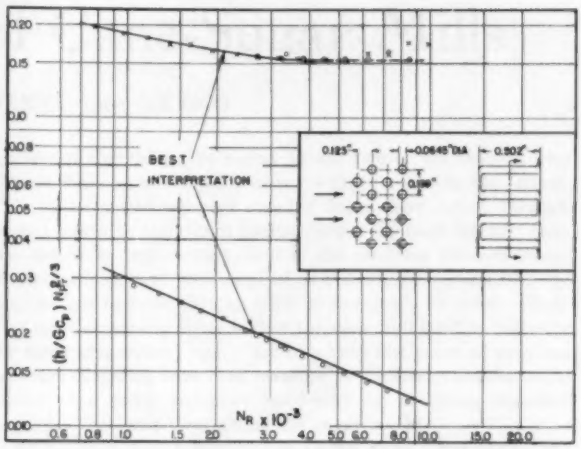
if this is correctly interpreted, Mr. Woolard has discovered a complete breakdown of the principles of dynamic and thermal similarity. When this is coupled with Mr. Woolard's remarks concerning the influence of plate spacing, one is tempted to question the test data, or the interpretation of the test data, to which he refers.

As part of the conclusions of the paper, the author mentioned that some of the difficulties encountered with in-line pins might be overcome by the use of a staggered-pin arrangement. Since the original presentation of the paper, a staggered-pin core has been tested. This core was constructed by methods described in Mr. Sven Holm's discussion, Figs. 12 and 13. A very good and clean pin-to-plate bond was attained. The test core was of multiple-tube construction, similar to those described in reference (8), and really represents the first successful attempt at building by production methods a pin-fin heat exchanger for high-temperature surface with an aerodynamically clean pin-to-plate bond. A summary of the test results is presented in Table 9 and the results are shown graphically in Fig. 18 of this closure.

The staggered-pin spacing is approximately $2D$ in the flow direction, and $3D$ in the transverse direction, so that this is a staggered version of PFS, Fig. 4, but of course built by production rather than shop methods. The data indicate no tendency toward laminar flow in the Reynolds-number range considered, but the surface whistled at Reynolds numbers above 5000. The heat-transfer performance is very high but so are the friction factors.

TABLE 9 SURFACE PF-4

N_R	$N_{St}N_{Pr}^{2/3}$	f	$N_{R(d)}$
800	0.0330	0.198	218
1000	0.0301	0.189	272
1200	0.0279	0.182	327
1500	0.0255	0.175	469
2000	0.0227	0.167	545
2500	0.0207	0.160	681
3000	0.0192	0.156	817
4000	0.0161	0.153	896
5000	0.0156	0.153	1362
6000	0.0144	0.153	1634
8000	0.0128	0.153	2179



$r_h = 0.00493$ $\quad A_f/A = 0.638$
$N_{R(d)}/N_R = 0.272$ $\quad \beta = 144$ sq ft/cu ft

o, hot-core test points
x, isothermal friction factors

FIG. 18 BASIC HEAT-TRANSFER AND FLOW-FRICTION DESIGN DATA, SURFACE PF-4

In conclusion, the author would again like to thank all of the participants in this rather extensive but enlightening discussion of pin-fin surfaces.

Grinding Capacities of Cane-Sugar Mills

By JOSE M. CABRERA,[1] MEXICO City, MEXICO

This paper summarizes milling equipment in actual operation as listed in the "Gilmore Manuals for 1951," for Cuba, Florida, Louisiana, and Hawaii, and for 1950, for Puerto Rico, Dominican Republic, and Haiti. Text with tabulations and graphs covers mill sizes, crusher types and arrangements, number of milling units in tandem, tonnage ground, sucrose in cane, fiber in cane, maceration employed; it includes various curves comparing the capacities and extraction by mill sizes, crushing arrangements, and other data pertinent to the task of grinding cane.

IT is hoped that this paper will be a positive contribution to attaining greater milling capacities by comparing what is being done in some countries with the same basic material; cane, mills, humans, and interchange of ideas in the profession of engineering. The findings have been confined to the data obtained from the "Gilmore Manuals for 1951" in Cuba, Hawaii, Louisiana, and Florida, and for 1950 in Puerto Rico, Dominican Republic, and Haiti. The resulting graphs, tables, and text are not intended to favor any one school of thought, but rather that they be used as a measure of each individual's effort to improve his work and surpass that of his neighbor.

The basic purpose of any industry or commerce is to derive profit for the direct participants: capital, labor, government, and the business itself via reinvestments in machinery or materials to expand it. Those countries having directed economies may require allowing some enterprises to exist at the expense of others in order to insure a production commensurate with the needs of the population of the country. However, the limit will be the price the consumer public is willing and able to pay; from then on subsidies have to make up the difference until the situation becomes acute and something is done about it.

Conditions existing in Mexico have led the author to believe that the cheapest remedy is to improve the industrial efficiency and thus reduce the unnecessary load being carried by the industrialist as distinguished from the opportunist whose existence depends on the subsidies and consequently weighs on capital, labor, government, business, and the consumer.

Everyone remembers the saying, "the fellow that hits first hits hardest." In a measure this applies to cane milling; get all you can at the first try which is in the mill room. For this reason milling capacity was selected as the topic. Nearly every engineer has had occasion to compare what has been done in one or more foreign countries and in many instances he will have been surprised to note the great differences in results obtained with the same factors of climate, mills, cane, humans. Such ideas we cannot help but repeat time and again.

Rise of the Sugar-Cane Industry

The sugar industry in America dates from a few years after the landing of Columbus; in a way it was one of the reasons why Europe was so anxious to discover a quick route to the Orient. This industry has traveled steadily from slave labor, through animal power to the steam engine during the past century, electric motors in this century, and in the past few years to steam turbines which have been applied as mill drives. Tomorrow we may see gas turbines driving mills on low-grade by-product fuel. The most surprising thing is that the cane mill itself is basically the same as a century ago. Yet in a very few years all previous concepts of milling have been revamped completely, thanks to the efforts of a small group of hard-working, studious, unselfish engineers scattered throughout the sugar world.

We all know how much we owe to the Scots, the Cubans, the French, the Dutch, the North and South Americans, who have contributed generously in time and effort, each in his day, each in his capacity. All these nationalities have assisted the world to increase its earnings and augment its food supply from sugar cane.

Factors Affecting Sugar-Mill Production

What then are the factors affecting mill capacity, assuming the sugarhouse can take all the cane juice the mill extracts? Briefly, for a given mill size, the following may be considered the real measure of capacity: Fiber in cane, premilling preparation, mill settings, speed of rolls, type and power of the drive, number and arrangement of units in the tandem, maceration, extraction, and the human element. (See Appendix.) A review of these will be of interest.

Fiber in Cane. This material is generally defined as the dry insoluble matter in clean cane to which, in the more progressive countries, the insoluble matter contained in the trash is added, although it is determined separately. Until the turn of the century, fiber primarily was considered from a fuel standpoint and as the cause of wear in the mills. Present practice is dependent upon good fiber determination for proper settings, and well it might, since cane entering the mills runs between 10 to 20 per cent fiber on cane. Bagasse from the last mill may run 30 to 50 per cent fiber, depending on the equipment and the efficiency with which it is handled. The average compressibility of fiber in terms of bagasse is between 75 and 85 lb per cu ft so it is readily understandable that it should affect capacity. Most milling formulas are directly or indirectly based on fiber.

Tonnage of fiber in unit time that can be ground by a unit of area of crushing rolls requires a special term, and in view of the numerous systems employed by the trade, the author decided to use an irregular unit that may be converted easily to equivalents used elsewhere. The "metric tons of fiber ground per 24-hr day per sq ft of single-roll area" is thus written MTFD/ sq ft SRA where SRA is as defined in (2) of the Conclusions.

A series of graphs, Figs. 1 to 20, demonstrate the great divergence in performance from one country to another in terms of fiber ground per square foot of roll area, irrespective of tandem sizes or mill arrangements, or number of mill rolls in the tandem. Fig. 21 shows capacity and extraction curves from the averages of 342 milling tandems, while Fig. 22 shows base capacity and extraction curves, giving nominal increase due to each 3-roller mill following crusher and 9RM. With the new horizons constantly being found for agricultural by-products, fiber will soon become more an economic factor and less important as a source of fuel.

Premilling Preparation. Under this heading is found the classical knifing, the more recent shredding, and the still more

[1] Owner, Manager, Firm of Jose M. Cabrera, Engineer. Member of Mexican Government Committee for the Reorganization of Mexican Sugar Industry. Mem. ASME.

Contributed by the Process Industries Division and presented at a joint session with the Materials Handling Division at the International Meeting, Mexico City,, Mexico, March 10–12, 1954, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, December 11, 1953. Paper No. 54—Mex-14.

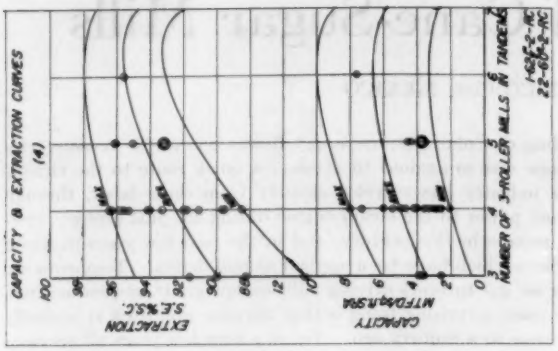


FIG. 4

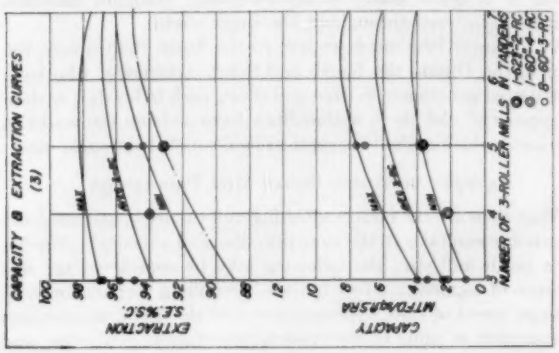


FIG. 3

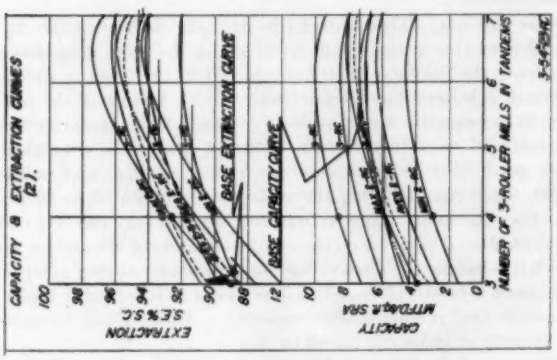


FIG. 2

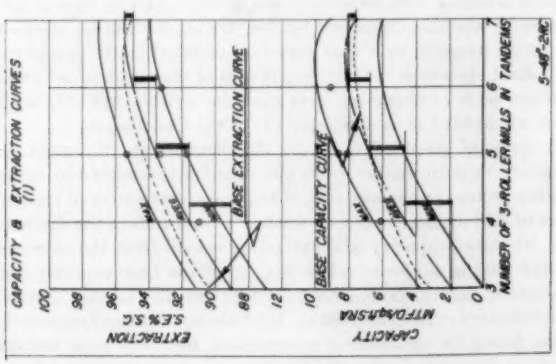


FIG. 1

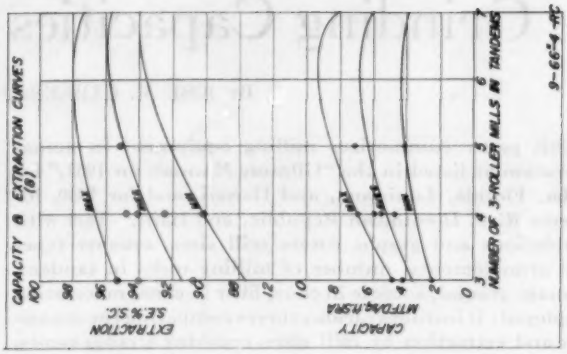


FIG. 8

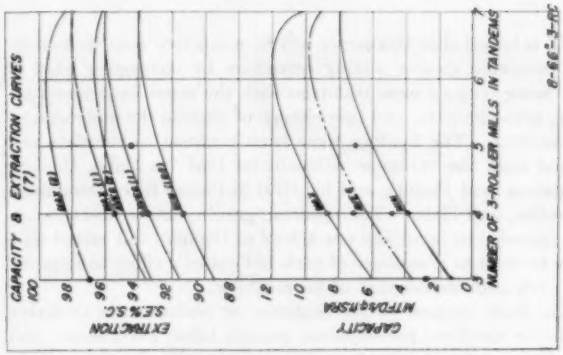


FIG. 7

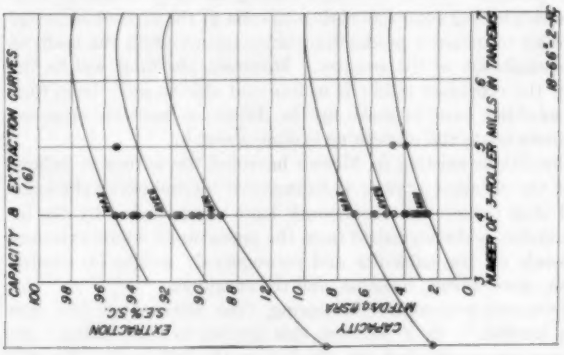
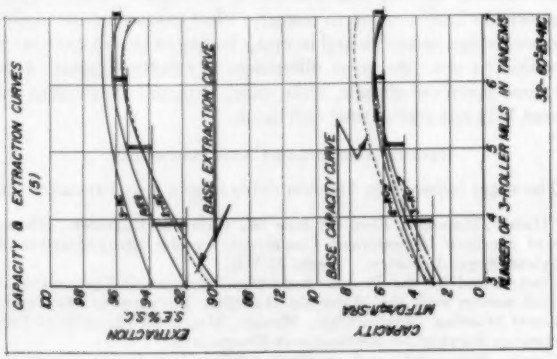


FIG. 6



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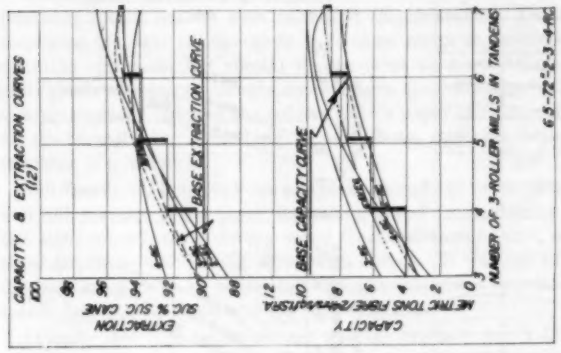

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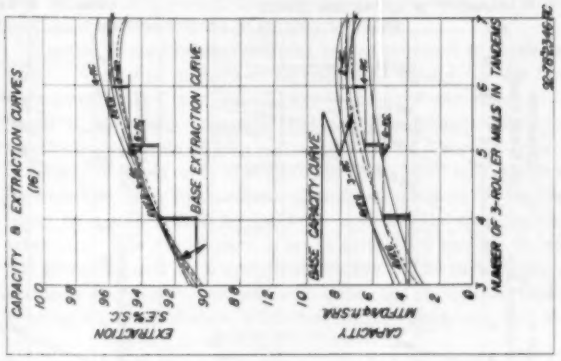


FIG. 16

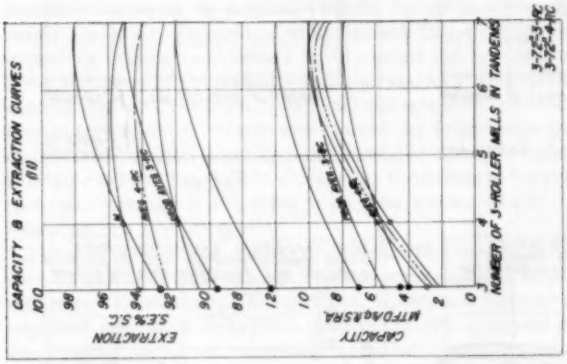


FIG. 11

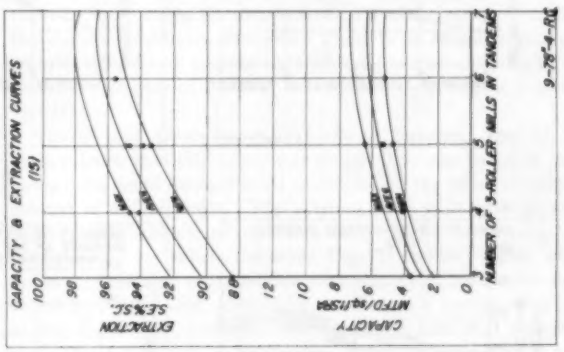


FIG. 15

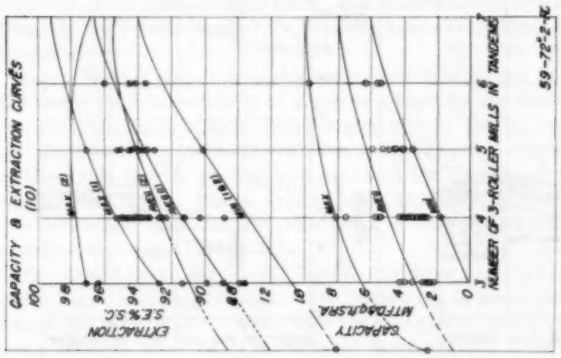
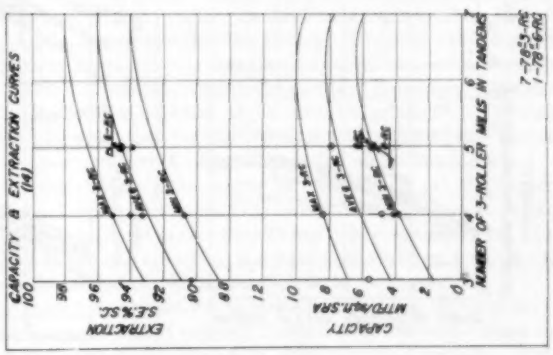


FIG. 10



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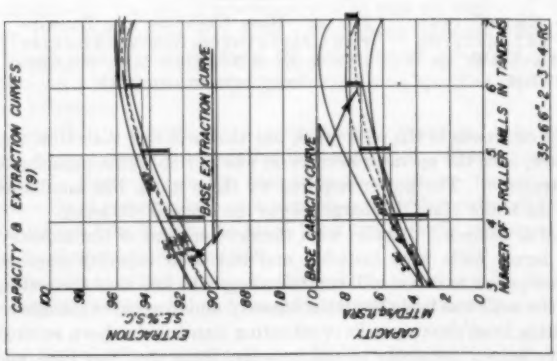

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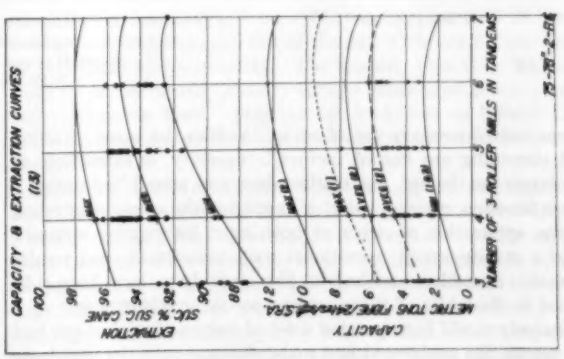

FIG. 13

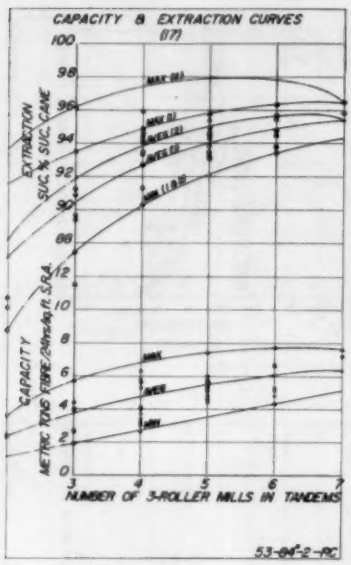

FIG. 17

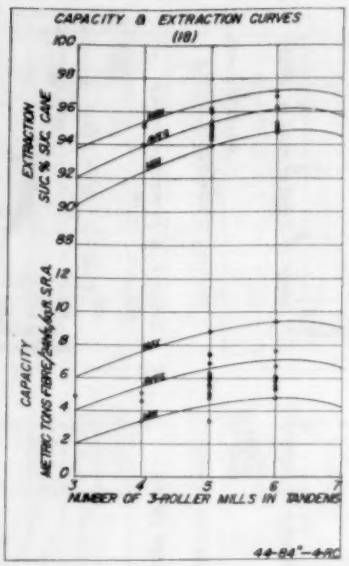

FIG. 18

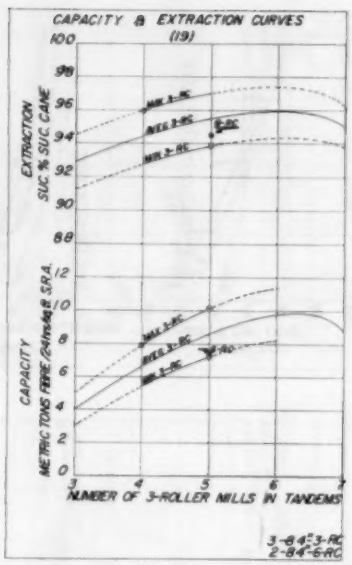

FIG. 19

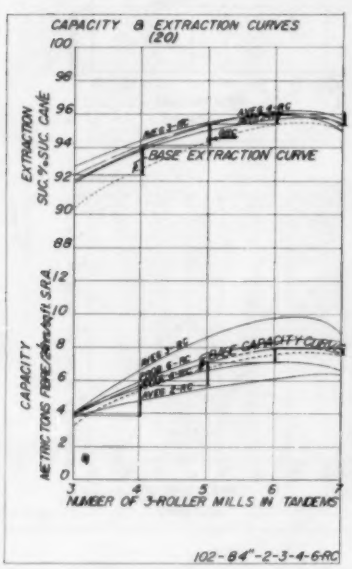

FIG. 20

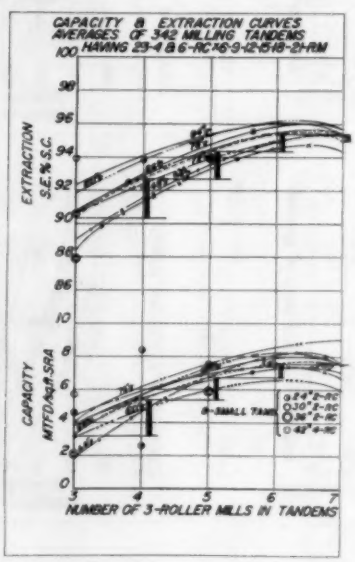
CAPACITY & EXTRACTION CURVES
AVERAGES OF 342 MILLING TANDEMS
HAVING 2-3-4 & 6-RC X 6-9-12-15-18-21-RM
EXTRACTION S.E. % S.C.
CAPACITY MTFD/sq ft S.R.A.
NUMBER OF 3-ROLLER MILLS IN TANDEMS

FIG. 21 CAPACITY AND EXTRACTION CURVES—AVERAGES OF 342 MILLING TANDEMS HAVING 2-3-4 AND 6-RC × 6-9-12-15-18-21-RM

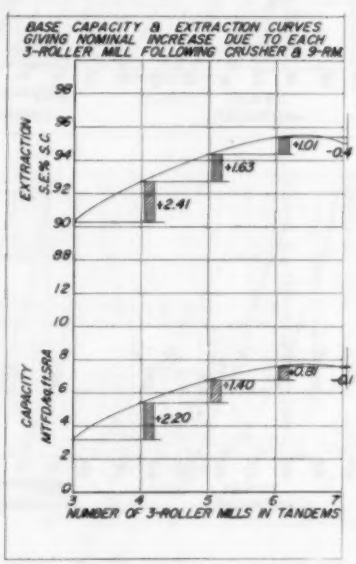

FIG. 22 BASE CAPACITY AND EXTRACTION CURVES GIVING NOMINAL INCREASE DUE TO EACH 3-ROLLER MILL FOLLOWING CRUSHER AND 9-RM

recent complementary use of water to clean the cane. Knifing and shredding are known factors in capacity, or extraction, or combinations thereof, yet neither does any actual "extracting." Their function is strictly that of preparing the cane for grinding so the application of either or both must be weighed seriously from a standpoint of operational cost, investment, and results desired. Shredders and knives, like straw hats, have been subjected to the whim of those anxious to try anything new.

Scarcely a mill has operated a set of knives which has not had to operate the equipment and make changes until the final location and peculiarities had been worked out empirically. Shredders are even more in the prima-donna class to judge from the conflicting trends in the past years, but they will find their time and place, once the operator decides on whether he wants capacity or extraction. The power required by these units, like auxiliaries in the boiler plant, is chargeable to the over-all efficiency.

Mill Settings. Parallel with the development of the industry, as larger mills were installed and increased capacity required more power to drive, mill engineers began to find that the setting of the mills was basic to attain capacity and to avoid breakages resulting from choking mills or vibrating stands. In short, settings have helped decidedly to reduce wear, breakages, lost time, and to improve capacity; they also have contributed to higher extractions.

Mill setting, while yet an art known to a selected few, is rapidly flowering into a science with rules and specifications. The associations of sugar technologists have done much to promote a scientific approach, but special thanks must be given to many individuals who have willingly made known their findings.[2] No attempt is made to cover the subject in this paper as each reader no doubt will have something to contribute, once his natural reticence is overcome.

Roll Speeds. There are three schools of thought as to the proper mill-roll speeds: Some favor increasing speed from crusher to last mill, some favor reducing speed from entrance to exit, and there are those who do not agree with either. It was not until the past score or so of years that thought was given to relative speeds between one mill and the following one.

Actually, the mill manufacturer was the one who had a little "black data book," and the mill operator was happy to take his recommendation on the subject. During the break of the sugar market some very experienced mill engineers found employment improving milling installations with a strict eye to economy. Some transferred from operation to design and vice versa and as a consequence the trade began to give mill-roll speeds due consideration in setting mills. Much was learned by interchange from man to man but little if any from mill to mill or society to society. It is quite evident that mill speeds are due to increase in the years to come especially if two-roller mills ever receive serious consideration in a new installation.

Types of Drives and Power Employed. The earliest drives were by means of slave labor which no longer exists in tropical countries. Centuries ago India, Africa, Korea, and Europe—all employed draft animals to drive mills as capacity increased and the physical size of the equipment exceeded human endurance on the treadmill. Animal power still is applied in many countries but not on any industrial scale. Water wheels came into use from the very beginning where conditions permitted their application.

James Watt invented the steam engine and thus opened up to the world the true potential of sugar as a food for the masses. By 1841 the reciprocating steam engine was by far the most popular drive for sugar mills. This new drive required reinforcing the mills and the gearing, and increased the capacity of the equipment. No set theory applied then as regards power to drive mills, and again the little black book of the mill and engine manufacturer reigned supreme.

The actual brake horsepower required to drive sugar mills is today quite empirical because of the fluctuation of steam pressure at the throttle and exhaust with the consequent variation in mean effective pressure. Considerable theorizing has failed to bring forth any set of rules that could govern actual horsepower for mill drives. The usual procedure was to pick an engine of the size that some other mill used for a given tonnage output. Though data are available on the power input per ton of cane, there yet is no figure on a fiber predicated for a specified capacity and extraction.

Less than a decade ago Louisiana again pioneered, and the first steam turbine was installed in a sugar mill. Frank Vought of Loula Factory took the first steps to determine torque and power-input requirements in a manner not heretofore employed. He did not take it from the black book but actually measured it on the equipment in operation.

The advent of the turbine to the task of the sugar mills raised considerable discussion and the author knows of several instances where the owners went to excessive expense to put in mill engines despite recommendations to employ turbines.

Mention should be made of the power required to drive cane mills. Numerous authors have published figures on the brake horsepower (bhp) per ton of cane. It is not the intent of this paper to start any argument but the fact remains that until figures are available on bhp per ton of fiber at a specific capacity in tons of fiber per sq ft of roll area or some other unit, there will be wide variations in criterion. Nearly 40 per cent of the power input to a mill is lost in the work done at the turnplate and trashbars; this would seem a point attractive enough to bring the two-roller mill back into consideration. The turbine drive is certainly an improvement and will bring new thought to bear on gearing, stands, pedestals, vibration, higher speed, and may even result in a new era in milling.

Other aspects of steam turbines is that they are the precursor of gas turbines and someday these may find wide application in the sugar industry where the molasses offers a ready fuel supply when the market conditions are right. There is no reason why a gas turbine should not operate part of the year on petroleum by-products and part of the year on molasses by-products properly carburized.

Number and Arrangements of Units in Tandem. One of the basic reasons that this paper was undertaken was to see if published data could be correlated to determine the effect of combinations of milling units. The original mills with vertical axes had two rolls, and a third roll was added to facilitate the return of the cane from one side to the other, Figs. 23 and 24. Later, when steam engines and water wheels were employed, the rolls were set on horizontal axes. Basically, a mill was a 3-roller affair until Mr. Krajewski came out with a crushing unit that achieved great popularity because it definitely was an effective tool to prepare the cane. From this we have the outgrowth of the Fulton 4-roller or double crusher, the 6-roller or triple crusher, and more recently, the so-called 3-roller crusher. There are rumors of quadruple crushers but none has been found in the records consulted.

This then was one of the great points of discussion: "Should a 2-roller crusher, a 4-roller, or a 6-roller be installed?" The manufacturer again took the lead in this matter and for years the plant engineer accepted the recommendation of the vendor.

Then also came the matter of mill sizes to get the capacity. Table 1 shows that out of 342 tandems under consideration 28 per cent are 84 in., 26 per cent are 78 in., 19 per cent are 72 in., a little over 10 per cent are 66 in., and 9 per cent are 60 in. long; in other words, 326 out of 342 tandems are 60 in. or larger.

Table 2 shows that 108 tandems had 14 rollers, 61 tandems had 17 rollers, 37 had 11 rollers. Table 3 gives the same figures from the standpoint of milling rolls, excluding crushing rolls. Tables 3 and 4 show 2-RC, 3-RC, 4-RC, and 6-RC arrangements, from which it is evident that 310 out of 342 tandems use the 2-RC or its derivatives; actually, 240 out of the 342 units are 2-RC. Only 32 installations report 3-RC. The graphs, Figs. 1 to 20, illustrate the variations in capacity for the same type of equipment and comparable fiber. This further leads one to believe that to obtain capacity, perhaps the most elusive factor is the human element.

Extraction and Maceration. These two terms are discussed simultaneously as they can hardly be separated when capacity is under consideration. Maceration is almost universally understood in terms of water employed per cent cane ground. Since the juice extracted is a function of the water added, it therefore must follow that if much work is done to take sucrose with water from the cane, then the capacity must suffer. However, since the product that a miller desires to recover is the sucrose in the cane and since extraction is a function of the water added, within limits, it would seem to be more logical to apply macera-

[2] A list of references of authors who have proposed milling formulas and derivations appears at the end of the paper. Incidentally, these authorities helped lay the foundation for rationalising the setting of mills.

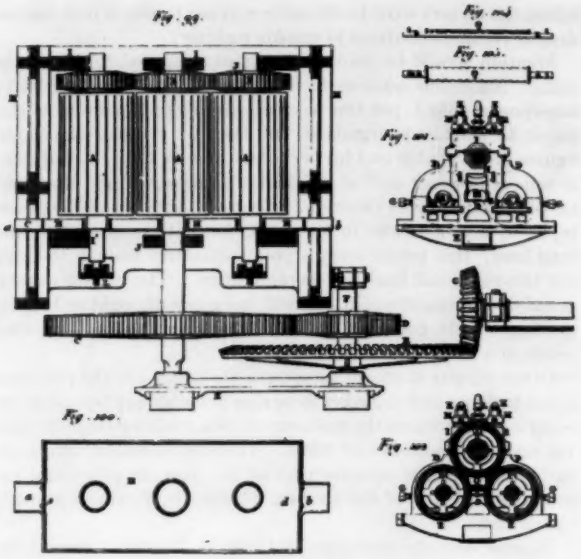

FIG. 23

settings and the careless attitude of helpers may defeat this by not tightening or adjusting a given unit to the proper point. The "let's do it later" or the "he's not so smart" attitude can do much to nullify the effect of good settings. There are some rare instances where a man may not wish to profit from published data because his pride will not let him admit that his method or theory has been at fault all along. There also is that inevitable character who wants to prove the factory wrong and who will change any piece of apparatus at the "drop of a hat."

Apparently the remedies applied by individuals to the milling problems did not follow any set pattern and so, consequently, each advanced in different directions with few attempting to co-ordinate data from various sources. The "Gilmore Manuals" initiated what may be considered the step in the proper direction. Data from mills all over the world were published so each subscriber could study in general terms what someone else was doing. No attempt was made to form an international committee to examine each phase of the industry based on the published capacity. We have the same arguments: "Would you put in a 2-roller or 3-roller crusher to get more capacity?" "Should we speed up a 4-ft mill or go to a 6-ft mill to meet the greater capacity?" "Shall we grind slow and reduce wear, or fast but make up roll cost on larger tonnages?" These and many other questions have come up from time to time.

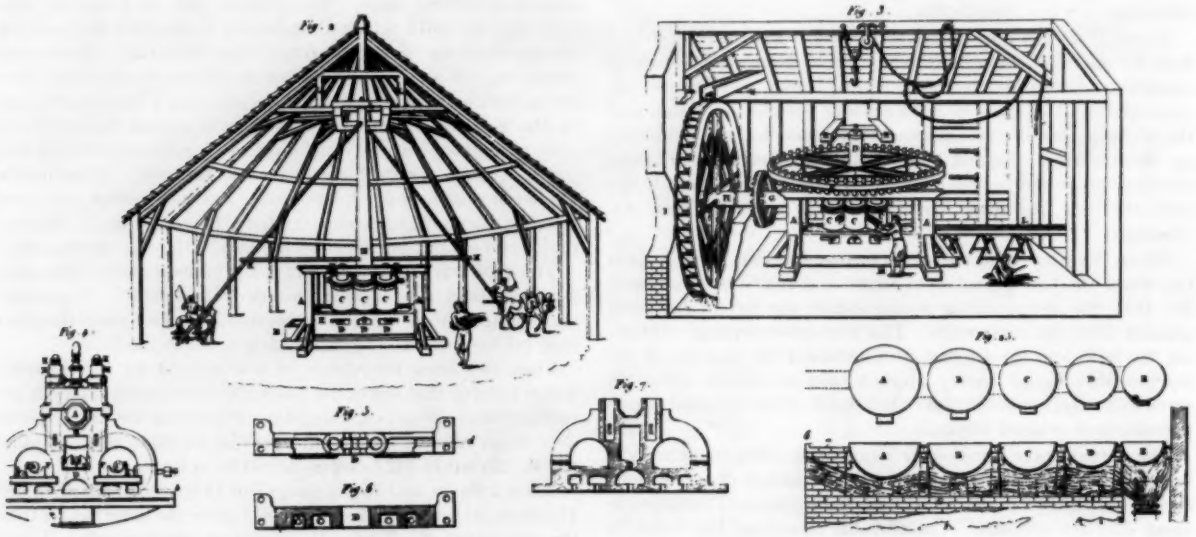

FIG. 24

tion water in proportion to the sucrose content of the cane rather than the basis of water per cent cane, or maceration per cent fiber. If maceration is applied in high ratio per cent of sucrose in cane, the extraction is good or excellent considering capacity figures; in some cases it even appears to help attain a higher unit capacity.

Human Element. This is undoubtedly the most unpredictable factor in milling cane. It begins in the field and affects cutting to proper sizes, topping, detrashing, loading; then in the "batey" its influence is found in the cane pile, tramp iron that goes into the carrier, loading of carrier, choking of knives; in the mill room at the crusher feed, evenness of feed to the mills, intermediate carriers, trash returns from elevator, cleaning up in mill troughs, and so on.

The critical point is in the mill settings. Here the engineer or the master mechanic, by theory or practice, has laid out good

The millowners took little or no interest in having their engineers travel to other mills outside the country to determine what they could learn. The investment in manpower and time was not justified because the owner himself perhaps did not grasp the importance of exchange of data.

Some years ago the Mexican mills needed to make changes to meet the growing market and, for lack of capital, tons of discarded machinery arrived in Mexico to saddle the industry with the same problem that had troubled the previous owners, i.e., "high cost of production on worn equipment." Yet statistics showed that Cuba kept in the lead despite worn equipment; thus it seemed to prove that the human element weighed more than expected. Some mill operators employed engineers trained in Cuba, who demonstrated that they were capable of grinding considerably more on the same worn equipment than had been possible prior to their arrival in Mexico.

TABLE 1 NUMBER OF TANDEMS BY COUNTRY AND ROLL LENGTHS

Country	Lengths of rolls in inches												Total Tandems
	24	26	30	36	42	48	54	60	66	72	78	84	
Hawaii								9	8	3	10	1	31
Puerto Rico Haiti & Dominican Republic	1	1	2	2	1	1	3	8	9	6	13	11	58
Louisiana & Florida	1					3		11	9	14	14	13	65
Cuba						1		4*	9	42	55	77	188
TOTAL	2	1	2	2	1	5	3	32	35	65	92	102	342

* Cuban tandem "Resolucion" rolls are 62½ in. long.

TABLE 2 TANDEMS BY TOTAL NUMBER OF ROLLS AND LENGTH OF ROLLS

Rolls in Tandem	Lengths of rolls in inches												Total Tandems
	24	26	30	36	42	48	54	60	66	72	78	84	
23											1	3	4
22											1	11	12
21											1	2	3
20						1		1		4	9	9	24
19									2		5	25	32
18				1				2	1		1	2	7
17					1		3		4	3	13	22	15	61
16								1	4	2	2	7	16
15			1				1		4		4	1	11
14		1				1	1	14	14	29	34	14	108
13					1				3	4	1	1	10
12								6	1	3	2		12
11		1			1		1	4	2	9	9	9	37
8									1	1		3	5
TOTAL	2	1	2	2	1	5	3	32	35	65	92	102	342

TABLE 3 TANDEMS BY NUMBER OF MILLING[a] ROLLS AND LENGTH OF ROLLS

Rolls in Tandem	Lengths of rolls in inches												Total Tandems
	24	26	30	36	42	48	54	60	66	72	78	84	
21											1	3	4
18						1		1		4	10	20	36
17											2		2
15			1	1		2		6	6	13	28	44	101
12	1	1				2	2	15	22	34	40	23	160
9	1		1	1	1		1	10	6	13	11	9	54
6									1	1		3	5
TOTAL	2	1	2	2	1	5	3	32	35	65	92	102	342

[a] Excluding rolls in all crushers or, where no crushers are reported, the first set of three rolls is excluded.

Everymill owner asks the builder, "How much can I grind per day on your mill?" Invariably the answer will be a conservative guess running between 60 and 70 per cent safe margin required, and the well-known phrase, "Your engineer will know; it depends on him." This paper was prepared *to see what someone else is doing with the same equipment in terms of fiber per day per square foot of roll area with equal extraction.*

TABLE 4 TANDEMS DISTRIBUTED BY LENGTHS AND NOMINAL ROLL DIAMETERS

Diameters**	Lengths of rolls in inches												Total
	24	26	30	36	42	48	54	60	66	72	78	84	
38*										1		13	14
37*								1			3	9	13
36*									2	7	15	61	85
35*								1		15	30	3	49
34*								6	5	32	42	15	100
33*								6	13	4			23
32*								12	15	5	2	1	35
31*								2		1			3
30*							2	4*					7
29*							1						1
28*					1		2						3
26*							1	1					2
23*				1	1								2
22*				1									2
20*		1				1							1
18*		1	1										2
Total	2	1	2	2	1	5	3	32	35	65	92	102	342

* Cuban tandem "Resolucion" rolls are 62½ in. long.
** Nominal diameters are the mean of diameter at top and bottom of grooves.

Conclusions

1 From the various curves presented in this paper it is evident that sucrose extraction per cent sucrose in cane is entirely independent of mill sizes as it is a function of settings of mills, maceration, number of milling units in a tandem, type of crusher employed, and other premilling operations, providing that nominal tonnage throughput is observed. In other words, extraction may vary, for any given mill size, depending on the tons of fiber per hr passed per square foot of mill-roll area; however, irrespective of the mill size itself the extraction will be fully parallel to the unit capacity in tonnage per unit area.

2 Capacity curves have been prepared using the basis of the term "single-roll area," i.e., the area corresponding to the 3-RM first in tandem but excluding the crushers. This area, here termed SRA, is computed from the nominal diameter (average diameter resulting from top of tooth and bottom of the groove figures) multiplied by the nominal length which is the distance between the flanges of the top roll of the first mill. In this manner the grooving arrangements and types are temporarily omitted since usual practice is quite general as to pitch and angle of grooving. These capacity curves reflect the results of crusher combinations and it should be noted that a 2-RC is exceeded by a 4-RC and this is in turn exceeded by a 3-RC as regards actual capacity and extraction, but that the 4-RC is much more economical in maceration water, pound for pound of recovered sugar, than is a 3-RC or even a 2-RC. The few installations of 6-RC units are not a sufficient basis on which to judge the performance;

TABLE 5 TANDEMS PER COUNTRY SUMMARIZED BY CRUSHER ARRANGEMENT

L- Roll	2-RC				3-RC				4-RC				6-RC				Total Tandems
	L	H	P	C	L	H	P	C	L	H	P	C	L	H	P	C	
84"	12	1	6	34			3		1		2	41				2	102
78"	10	10	11	44	3		2	2	1			8				1	92
72"	11	3	5	40	2		1		1			2					65
66"	7	4	2	5		4	4		2		3	4					35
60"	9	4	6	4*	2	5	1				1						32
54"	-	-	2	-	-	-	1	-	-	-	-	-	-	-	-	-	3
48"	3	-	1	1	-	-	-	-	-	-	-	-	-	-	-	-	5
42"	-	-	-	-	-	-	-	-	-	-	1	-	-	-	-	-	1
36"	-	-	2	-	-	-	-	-	-	-	-	-	-	-	-	-	2
30"	-	-	1	-	-	-	1	-	-	-	-	-	-	-	-	-	2
26"	-	-	-	-	-	-	1	-	-	-	-	-	-	-	-	-	1
24"	1	-	1	-	-	-	-	-	-	-	-	-	-	-	-	-	2
TOTAL	53	22	37	128	7	9	14	2	5		7	55				3	342

* Cuban tandem "Resolucion" rolls are 62½ in. long.

TABLE 6 TANDEMS SUMMARIZED BY CRUSHER ARRANGEMENTS

L- Roll	2RC	3RC	4RC	6RC	Total Tandems
84"	53	3	44	2	102
78"	75	7	9	1	92
72"	59	3	3	0	65
66"	18	8	9	0	35
60"	23*	8	1	0	32
54"	2	1	0	0	3
48"	5	0	0	0	5
42"	0	0	1	0	1
36"	2	0	0	0	2
30"	1	1	0	0	2
26"	0	1	0	0	1
24"	2	0	0	0	2
TOTAL	240	32	67	3	342

* Cuban tandem "Resolucion" rolls are 62½ in. long.

the good showing in one case could give a tendency to overestimate its potentials.

3 Base curves have been derived from the general average of the performance of each-size mill listed from 48-in-long rolls on up. The mills having rolls 24 in., 30 in., 42 in. are so limited in number that only points were plotted for the sake of comparison. Contrary to anticipated expectations, the capacity did not increase materially with increase in mill and roll sizes; the base curve is fairly comparable to the averages of the curves for each mill size irrespective of milling setups or crusher arrangements. Using the averages of each mill arrangement and each crusher arrangement, a series of increment factors have been shown by a shaded area. Likewise, since the base curves are averages of performance obtained under a standard practice, increment factors also were obtained from which it is evident that a 9-roller mill having knives and a crusher (2-RC, 3-RC, or 4-RC) is capable of grinding under average conditions 3 metric tons of fiber per sq ft of single-roll area per 24-hr day, which at a 12½ per cent fiber in cane represents a grinding capacity of 24 metric tons of cane per sq ft SRA. From data in Table 7 this can be converted into capacity for any given size of mill and roll.

A close inspection of the "maximum" results obtained, as compared to the base curves, will show that in actual practice it is, and has been, consistently possible to attain a maximum performance ranging from 100 to 200 per cent of the increased extraction factor, and from 200 to 400 per cent of the increased capacity factor, when a mill is added to an existing installation presently performing at or about the base-curve ratios. Therefore it is on this that the author bases his opinion that other engineers can improve their milling work by striving to equal the best.

Poor work has no definite range so minimum limits are not set. The plotting of the curves has been done from the actual reported data submitted by 342 tandems and does not make any allowance for those cases where the capacity was limited by factors other than the tonnage that the mill could handle. Also, where a sugarhouse is small and the milling capacity has to be limited, certainly a higher extraction should be obtainable than otherwise.

4 Time has not permitted the plotting of curves based on a "milling-roll area" instead of the "single-roll area." This would take into account all the rolls actually milling but excluding the crushing rolls, i.e., excluding all 2-RC, 4-RC, 6-RC units, and any 3-RC where these units are employed. This being the real milling area in a tandem, certainly such curves, in general, would reveal the effect of the different arrangements of crushing rolls.

5 Another series of curves that could be drawn is the total roll-area basis wherein all rolls would be taken into account. This would allow for an over-all capacity consideration and the variations in extraction, capacity, maceration, and so on, could again be analyzed in relation to settings, human element, and the like, providing comparable fiber per cent cane is ground and a co-ordinated effort is made to obtain data specifically prepared for this purpose. Some preliminary curves, not included in this paper, seem to bear out that capacity is decidedly not a function of mill and roll sizes.

6 To establish a further basis for comparison the term "metric

TABLE 7 MILL-ROLL NOMINAL SIZES AND SINGLE-ROLL AREA

Nominal Roll Ø	Nominal Circumf"	Lengths of rolls in inches 87"	84"	78"	72"	66"	60"	54"
44	138.2		80.6					
42	131.9		76.9					
41	128.7		75.1					
40	125.6		73.2					
39	122.5		71.5	66.4				
38-1/2	120.9		70.5	65.4				
38	119.4		69.6	64.6				
37-1/2	117.8		68.7	63.8				
37	116.2		67.8	62.9				
36-1/2	114.6		66.85	62.1				
36	113.1	68.4	66.0	61.3	56.6	51.8		
35-1/2	111.5		65.0	60.4	55.7			
35	109.0		64.14	59.45	54.9		45.78	
34-1/2	108.4		63.2	58.7	54.2	49.7	45.2	
34	106.8		62.4	57.86	53.3	47.85	44.5	
33-1/2	105.2		61.3	56.98	52.6	48.23	43.85	
33	102.7		60.5	56.2	52.2	47.5	43.2	38.9
32-1/2	102.1			55.3	50.11	46.83	42.56	38.3
32	100.5			54.14	50.04	46.05	41.88	37.68
31-1/2	98.9			53.6	49.45	45.3	41.2	37.1
31	97.4			52.55	48.7	44.64	40.6	36.5
30-1/2	95.6			51.75		43.81	39.75	35.85
30	94.3			51.07		43.25	39.8	35.3
29-1/2	92.7						38.64	34.75
29	91.15						38.00	34.18
28-1/2	89.55						37.3	33.6
28	88.0				44.4	40.33	36.66	33.0
27-1/2	86.4							32.4
27	84.8						35.3	31.8
26-1/2	83.25							31.25
26	81.7					37.4		30.64
	LR Length Ratio:	1.000	0.9285	0.8570	0.7859	0.7141	0.6425	
	Area Factor K	0.5833	0.5416	0.499	0.4583	0.4166	0.3749	

NOTE: SRA is single-roll area in square feet equal to $3.1416 \times L \times D \div 144$. SRA equals $K \times D$ where K is area factor equal to $L \times 3.1416 \div 144$ or $0.006944 \times L$.

TABLE 8 MILL-ROLL NOMINAL SIZES AND SINGLE-ROLL AREA

Nominal Roll Ø	Nominal Circumf"	Lengths of rolls in inches 48"	42"	36"	30"	26"	24"	22"
31	97.4	32.48						
30-1/2	95.6	31.87						
30	94.3	31.44		23.56				
29-1/2	92.7	30.9		23.17				
29	91.15	30.58		22.78				
28-1/2	89.55	29.85		22.38				
28	88.0	29.33		22.0				
27-1/2	86.4	28.8		21.6				
27	84.8	28.45	24.72	21.2				
26-1/2	83.25	27.75		20.1				
26	81.7	27.23		20.45				
25-1/2	80.11	26.7		20.0				
25	78.56	26.2		19.62				
24-1/2								
24	75.4	25.15		18.86				
23-1/2								
23	72.25	24.09		18.06	15.50			
22-1/2	70.7	23.8		17.66	14.73			
22	69.15		20.18		14.40			
21-1/2								
21	65.97			16.5	13.74	11.91	10.99	
20-1/2								
20	62.83			15.7	13.23	11.36	10.48	
19-1/2								
19	59.70							
18-1/2	58.13							
18	56.55				11.81	10.31	9 .45	
17-1/2								
17	53.41							8.43
16-1/2								
16	50.26							
Area Factor K		0.3333	0.2917	0.2499	0.2083	0.1806	0.1666	0.1578
LR Length Ratio:		0.5710	0.5000	0.4285	0.3571	0.3084	0.2858	0.1847

NOTE: SRA is single-roll area in square feet equal to $3.1416 \times L \times D \div 144$. SRA equals $K \times D$, where K is area factor equal to $L \times 3.1416 \div 144$ or $0.006944 \times L$.

tons" of fiber per 24-hr day per "hundred" square feet of milling-roll area may be considered. This would place all capacity and extraction figures on a trend basis rather than any specific case. Preliminary work done on such figures has confirmed the impression that capacity and extraction are not functions of size of mills or rolls.

7 A curve of metric tons of fiber per 24-hr day per 100 sq ft of total roll area is also to be developed as supplementary to the series mentioned in the preceding paragraph.

8 The true measure of the effectiveness of maceration is yet to be gaged in terms of the pounds of water per pound of sugar recovered over and above the obtainable extraction intrinsic to the nature of milling cane. Undoubtedly, the "absolute juice" has a direct relation with the obtainable extraction; then, in addition, the maceration water is instrumental in obtaining greater extraction. Surprisingly, a study of the figures prepared by the author, but not yet published, seems to indicate that maceration, in some specific conditions, does contribute to a large increase in capacity. Conversely, in some cases it is evident that maceration is being wasted. There are numerous instances where, for a given capacity of the same MTFD/sq ft SRA and a comparable extraction, say, 94.3 to 94.6, the maceration can vary from 175 per cent water on sucrose in cane to 275 per cent, or in other cases, from 93 per cent water on sucrose in cane to 235 per cent. This would indicate that in some cases the *added half point of extraction requires nearly 90 per cent of the maceration water employed.*

Referring again to the author's suggestion of computing maceration water not in terms of per cent on cane but rather in terms of water per cent sucrose in cane, it is evident from the figures under consideration that some form must be devised to study the effect of such computation. For the moment let it suffice to say that extraction figures indicate considerable improvement to be found where the water employed ranges from 150 to 250 per cent water on sucrose in cane. What would be the result at 300 or even 400 per cent?

9 Temperature, pressure, and so on, are other factors to be considered but not undertaken in this paper.

It is suggested that engineers interested in further development of studies along this line should address their local engineering society to enlist co-operation to make available data in a society-to-society plan through specialized committees. In this manner there would be a continuity of ideas and information, otherwise not possible by individual effort, despite the best of intentions. Then also by a co-ordinated effort through societies the cost of carrying on such work can be shared by all members and sponsored by cane millers, cane-mill builders, government agencies, and *cane growers*, who beyond any doubt are the most favored ones since cane is liquidated nearly everywhere in the world on a basis of sugar.

There are 1360 milling tandems producing a total of 22,000,000 metric tons of cane sugar so the direct results of a consolidated effort to study this problem would net millions of dollars' savings and billions in business in the next 10 years.

No figures have been obtainable on the Mexican sugar industry because as yet there is a certain reluctance on the part of the millers to publish data on their equipment. With the exception of Ingenio El Potrero, Ingenio San Cristóbal, El Modelo, Los Mochis, and three or four others, there is a singular lack of interest in the exchange of data to improve milling conditions.

A future paper of greater scope would be prepared if data were obtainable from other cane-milling areas and if there is any interest in the subject. Further fields can be covered by the consideration of these curves for determining power, speed, torque, and other aspects of the art of milling cane.

Acknowledgment

Special thanks are here given to Mr. Fred I. Meyer of the "Gilmore Manuals," for permission to publish these data and to the authors and sources given in the list of references which follows, from whom the present author has drawn generously.

References

The following abbreviations will be used to identify publications and societies mentioned in references:

ISJ = *International Sugar Journal*
FAS = Facts About Sugar
ISSCT = International Society of Sugar Cane Technologists
ATAC = Asociacion de Tecnicos Azucareros de Cuba
I Mech E = The Institution of Mechanical Engineers.

1 Noel Deerr, ISJ, vol. 34, October, 1932, pp. 389–392; ISJ, vol. 48, 1946, p. 234.
2 P. H. Parr, ISJ, vol. 24, 1922, pp. 251–257, 299–304; ISJ, vol. 37, 1935, pp. 355–356.
3 T. Wishart, ISJ, vol. 22, 1920, pp. 625–629.
4 T. H. P. Herriot, refer to (27).
5 G. Brooke, refer to (27).
6 J. Cerban, refer to (27).
7 H. Walker, *Sugar News*, vol. 8, 1928.
8 F. A. Lopez Ferrer, ATAC, 1939, p. 235.
9 M. Ch. Varona, ISJ, vol. 39, no. 462, 1937, pp. 222–226; ISJ, vol. 43, 1941, pp. 17–18.
10 L. A. Tromp, op. cit., "Machinery and Equipment," p. 165.
11 W. Kilpatrick, I Mech E, vol. 125, 1933, p. 615.
12 H. J. B. Scharnberg, FAS, vol. 23, 1928, pp. 10–13.
13 J. R. Mayo, ATAC Proc., 1938, pp. 88–90.
14 E. W. Kopke, FAS, vol. 27, 1932, pp. 57–59.
15 J. G. Salinas, ATAC Proc., 1939, p. 133.
16 F. Maxwell, *Modern Milling Sugar Cane*, p. 208.
17 Camp and Leonard, ISSCT, 1927, second Congress, p. 171; ATAC, Proc., 1931.
18 Thomas Lowe, *Sugar*, August, 1949.
19 Juan J. Pena, ATAC Proc., 1938.
20 J. C. Gonzales Mais, ATAC, XXIV, p. 383.
21 Jeronimo Diaz Compain, ATAC Proc., 1939, p. 39; ATAC, XXI, p. 279.
22 G. H. W. Barnhardt, AHST, 1925.
23 Segundo Perez Alonso, ATAC, XXV° Proc., p. 329; ATAC, XXII, p. 243.
24 L. R. Bliss, ATAC, XXII, p. 249.
25 Edgardo F. Gonzalez, ATAC, XXII, p. 273.
26 N. A. Helmer, FAS, vol. 17, 1923.
27 T. S. Ebro, *Sugar News*, vol. 13, no. 21, 1932, pp. 77–79.
28 E. M. Copp, FAS, vol. 23, 1928, p. 280; FAS, vol. 23, 1928, p. 326; FAS, vol. 23, 1928, p. 350.
29 Pieter Honig, ATAC, XXV, p. 372.
30 Ricardo Lehky, ISJ, vol. 39, 1937, pp. 137–141.
31 C. C. Savoie and F. A. Vought, ISSCT, 7th Congress, p. 710.
32 E. L. Dennis, ISSCT, 6th Congress, p. 1019.

Appendix

1 Equipment in the sugar factory whose mechanical conditions may constitute a factor limiting the mill capacity: heaters, evaporators, pans, centrifugals or other bottlenecks. The minimums of the curves are probably due to these.

2 Raw material cane: fiber in cane, availability, class, size cut, etc.

3 Human element: capital, supervision, labor.

4 Imbibition and maceration practiced.

5 Cane carriers, loading, and auxiliaries.

6 Preliminary preparation: knives, shredders, washing.

7 Crushing equipment, viz., 2,3,4,6-roller.

8 Milling equipment, viz., 3, 6, 9, etc. roller.

9 Mill roll sizes, i.e., diameter, length, grooving both axial and peripheral, including Messchaert.

10 Gearing design and condition.

11 Peripheral speeds and interrelation of speed between one mill and the next.

12 Settings: openings, discharge, trash-bar and scraper designed clearance, etc.

13 Intermediate carriers.

14 Housing design: with or without kingbolt, etc. inclined, etc.

15 Accumulators and hydraulic rams, sizes, and conditions.

16 Mill drives and available power, throttle and exhaust pressure of engines and turbines; voltage, current, power factor of electric motors, etc.

17 Trash-elevator dispositions.

18 Foundations and bed plates.

19 Normal maintenance and repairs.

20 Replacement and substitution cost due to higher capacity desired.

Discussion

A. P. Leonard.[3] In the words of the author, "This paper was prepared to see what someone else is doing with the same equipment, in terms of fiber per day per square foot of roll area with equal extraction."

As the result of voluminous work, the author presents graphically the capacity and extraction obtained in the commercial working of 342 cane-milling tandems.

The mills are of various periods, a few representing the last word in development by the leading mill manufacturers; many are of the vintage of the 1915 to 1930 era, and others are as much as 50 years or more old.

Many of the older mills have serious limitations inherent in their design. For example, higher speeds and therefore capacities are not feasible; hydraulic loading may be definitely limited because of inadequate bearings or lack of water cooling; the strength provided in the mills, gearing, and other parts may be insufficient for modern capacities with their higher power requirements, and so on.

As an example of the capabilities of a modern mill, may be cited the case of the 36-in. × 84-in. — 18-roll and-crusher tandem in use at Ingenio San Cristóbal, Mexico, for the operation of which the writer is responsible. This sugar factory is representative of the most progressive management in Mexico.

For the past 16 weeks of the present grinding season, this tandem has been grinding at the rate of 222 to 235 metric tons of cane per hr. It was set for 228 tons per hr, equivalent to 5500 metric tons per day (6000 short tons) and this figure will be used in comparing the work with the curves given in the paper. This mill corresponds to curves in Fig. 17. From the capacity curve marked maximum it will be seen that the largest capacity recorded for a tandem of 6 mills and crusher is 7.8 metric tons of fiber per 24 hr per sq ft of single-roll surface.

The San Cristóbal tandem is grinding 5500 metric tons of cane, of 15.327 per cent fiber to date (March 8) or 840 tons of fiber per 24 hr. The single-roll surface is 36 in. diam × 84 in. long or 66 sq ft by the author's Table 7. The metric tons of fiber per square foot are then 12.7 or over 60 per cent greater than the maximum of the curves in Fig. 17.

The sucrose extraction for this tandem to date (March 8) is 93.39 per cent, or referred to a $12^1/_2$ per cent fiber basis, is 94.109.

For both tandems the average sucrose in bagasse is 2.07 per cent to date and the sucrose per cent fiber is 4.378 to date. As the 36-in. × 84-in. tandem is doing slightly better work than the smaller tandem, the sucrose in bagasse and sucrose per cent fiber of the 36-in. × 84-in. tandem are somewhat better than the figures cited.

This mill is of the most modern design and has been improved in many details since its installation.

[3] Consulting Engineer, Sugar Industry, Sierra Gorda #405, Mexico City, Mexico. Mem. ASME.

Operating at what seems to be an extremely high rate, we find some limitations of capacity. It is felt that further modifications will enable even better extraction at the same capacity.

The question arises, what is the ultimate tonnage we may expect to be able to grind with an extraction economically justifiable?

There is no definite answer to this question; but it is the writer's belief that by making the necessary modifications a considerable increase is definitely in sight.

Based on the foregoing results, it appears that maximum mill-grinding capacities depend on the following factors:

1 An adequate design on the part of the manufacturer is basic.

2 Modifications to be made as necessary to conform to conditions existing at the sugar factory in question.

3 Carefully worked out mill settings.

4 Good maintenance and operation.

Author's Closure

Mr. Leonard does not consider the author's assumption that his four points are basic in any well-conducted mill, such as is to be expected in the countries surveyed (nearly 75 per cent of the production of America) and that the purpose of the paper was to explore possibilities of still further improvement in production and production methods. Unfortunately, a detailed case analysis is beyond the scope of this closure as it would require another paper almost as long as that under discussion. Let us, nevertheless, consider the four points raised by Mr. Leonard.

1 Adequate design must be sought in any new additions or new tandems, but in operating an existing tandem the engineer has no other recourse than to make the best of what he has. One of the purposes of this paper is to bring to the attention of the mill operator that, within limits, it is possible to do still more with what is already in operation. The tandems surveyed having completely new equipment of recent installation do not exceed 10 per cent, though no special effort was made to keep track of the new installations.

2 Modifications to conform to existing conditions apply to new equipment and particularly also to used equipment that is subject to changes in roll diameters, journal sizes, speed of gearing, reinforcing of stands and housings, modern accumulators, and top caps; in fact, dozens of other factors warrant consideration. Figs. 23 and 24 in Manuels Roret "Nuveau Manuel Complete du Fabricant et du Raffineur de Sucre de Cannes. . .", published in 1841, over one hundred years ago, show that cane mills were not much different in operating design from what they are now, though the *technique* has improved considerably in the actual work obtained out of the same-size equipment. Even so far back when the feed grooves were used, turnplates were grooved front and back. In fact, these mills had practically all that the modern mills have, except hydraulic top caps, accumulators, forced feed rolls, etc.

3 The fact that San Cristóbal grinds 60 per cent more than the maximum of the curve Fig. 17, and the fact that the San Cristóbal smaller tandem is doing work only slightly less good than the larger, despite the difference in age and design, leads one to believe that mill-setting technique bears a direct relation to the general improvements that can be made to existing tandems. This factor alone, if adverse, can nullify the manufacturer's best design and the other considerations.

4 Good maintenance and operation mentioned as limiting capacity, are, most likely, dependent on the economics and policy followed by the company owning the plantation and the mill.

The author's paper therefore provides empirical data for study, reclassification, and further discussion for application as contributions to the ultimate development of milling formulas, correlated with mill-setting formula for the general good of the industry.

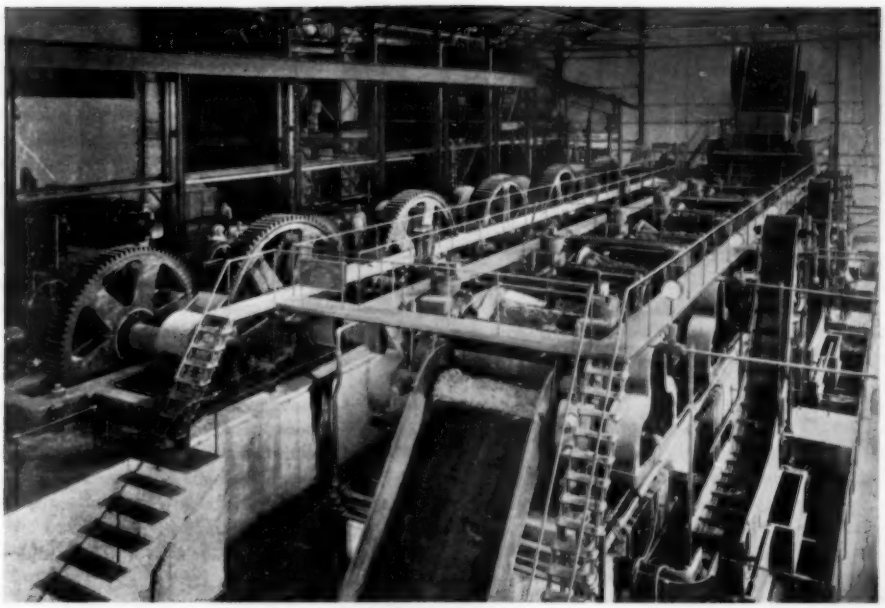

FIG. 1 TANDEM GRINDING MILLS

Conveying Equipment in Cane-Sugar Mills

BY C. CONRAD RINELLI,[1] MILWAUKEE, WIS.

This paper is a discussion of the value and function of conveyers in cane-sugar mills. It further deals with some of the basic factors and considerations involved in their operation. A solution of several problems of mechanical handling also is offered. Considerable emphasis has been placed on the application of chains, slats, sprockets, and other moving parts of the main conveyers and elevators in various sizes of cane-sugar mills.

INTRODUCTION

IN this paper the value, application, and function of conveyers, and some of the basic factors which are involved in the solution of mechanical handling problems of cane, bagasse, and sugar in the sugar mill will be discussed. Since chains, slats, sprockets, and other moving parts are the most important factors in the mechanical handling equipment, considerable emphasis will be placed on their application in various sizes of sugar mills.

IMPORTANCE OF CONVEYERS IN THE CANE-SUGAR INDUSTRY

The principal source of commercial sugar is sugar cane. It is mainly grown under semitropical conditions. About 12 to 14 per cent of the total weight of cane is turned into sugar and molasses. The total world production of cane sugar in 1953 was about 36.6 million tons. This represents about 375 million tons of cane weighing approximately 22 lb per cu ft which had to be handled and rehandled through the sugar factories in order to produce this volume of sugar. Without conveyers this production would have been impossible.

When one considers that the No. 4 (world) exchange spot price in September, 1953, was 3.35 cents per lb for raw sugar he will begin to realize that sugar making is one of the world's most efficient industries. Conveyers play a very important part in effecting cost reductions that make the low price possible.

The grinding season varies in different countries. It lasts about 100 to 140 days in the West Indies but in Colombia and Peru, for instance, the grinding goes on all year round. Once started, a mill grinds 24 hr per day. Mills range in width from 24 to 84 in. and occasionally wider. Approximately 1 short ton of cane is needed to produce 250 lb of raw sugar. To produce 200,000 bags of 300 lb each or 60,000,000 lb of raw sugar, means grinding 240,000 tons of cane per period of 100 days or 2400 tons per day. This is a lot of cane to handle, yet it is ground by a medium-sized mill. Central Jaronu in Cuba is reputed to be one of the largest in the world with grinding capacity of 12,500 tons per day with 3 tandems.

Sugar mills grinding up to 1000 tons per day of cane in one tandem, that is, a series of grinding mills in line, Fig. 1, may be considered small. One grinding 1000 to 3000 tons may be considered medium size and 3000 to 5000 may be considered large. As many as three tandems may be placed side by side, each grinding from 3 to 5 thousand tons per day.

Sugar making, aside from the agricultural phase, is divided into two general divisions: (1) The mechanical process of han-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, November 10, 1953. Paper No. 54—Mex-2.

dling the large volume of raw material, that is, the cane and waste matter, and (2) the chemical process of changing the sugar solution to refined sugar. A third division might be added, that of handling the sugar from the mill to the ultimate consumer.

It is the first and third divisions where the mechanical problem of handling raw and waste material, as well as sugar crystals in the various stages of refining and shipping, is solved by the use of mechanical conveyers. They provide for the integrated flow of cane to and through the milling process and then the flow of bagasse to the boilers to be burned to furnish power for the mill. (Bagasse is the mass that is left after all the juice possible has been pressed out.) In the refinery, conveyers and elevators handle raw and refined sugars in the various stages.

Any stoppage in the conveyer system shuts down some of the operations immediately and in a short time paralyzes all of the sugar-making process. Stoppages are the chief engineer's nightmares. Most important, however, is the high cost of shutdowns which in some cases amount to as much as $1000 per half hour.

Conveyers in the sugar mills, of course, must be tailored to fit the various sizes of mills, from the smaller ones making sugar cake, to the larger sugar mills which will grind from 12 to 15 thousand tons of cane in a 24-hr day. In his experience the author has found that all small sugar mills have the desire to become large. This is because the added capacity provides a more than proportionate return on the capital investment. For instance, Ingenio San Cristóbal in Mexico was designed originally to grind about 500 tons of cane per day and today is planning for a grinding rate of about 9000 tons per day. It is the job of the mechanical engineer, in so far as possible, to use standard designs of conveying units which afford smooth uninterrupted operation as well as give flexibility for possible future increases in production and in the handling of the finished product.

Cane-Harvesting, Conveying, and Cleaning in Hawaii

In Hawaii cane is harvested differently from any other place. To keep down harvesting costs, cane is not cut, topped, and cleaned, but pulled out of the ground with grapples. Roots, dirt, rock, leaves, tassels, and even the proverbial kitchen sink are sometimes found among the trash removed before milling. Therefore, in addition to handling of cane by conveyers, the trash must be removed by a cleaning process prior to crushing. For this purpose special cleaning apparatus is used as well as conveyers. These must be rugged in their construction. While the process differs slightly at various mills, in general it is as follows.

Harvested cane is dumped from cars to a receiving slat-type carrier which is inclined at the end to discharge into an unscrambling conveyer for the removal of rocks and dirt. This conveyer is on a sharp incline, so that rocks can roll back onto a rock conveyer while the cane is carried forward. Near the top of this conveyer is a spreading drum that spreads the cane and allows more of the debris to fall loose. The dirt and rocks cascading down create hammering and grinding compounds that are hard on chains. This conveyer has pipes mounted on chains at intervals to permit the dirt to fall through onto a dirt conveyer which carries it to trucks to be spread on the fields again.

From the unscrambling conveyer, the cane is either passed to a cascading washing conveyer and combing drums for trash extraction or to a sink and float tank for washing. In this tank the cane floats, whereas the debris that is left sinks. This idea was borrowed from the mining industry. From here the cane passes to a clean cane conveyer which is a conventional-type cane carrier. Sometimes it is made up in two sections—a horizontal and an inclined section. Sometimes there is only an inclined section, but in either case the incline is for the purpose of getting sufficient height to discharge into the chute over the crusher rolls or the shredder.

Cane Carriers

The cane-carrier system is the first group of conveyers to handle the cane after it is received from the fields. The purpose of the cane-carrier system is to bring an even volume of cane to the crushing rolls. Various feeder-table and feeder-carrier arrangements are used to supply the main cane carrier.

In the small to medium-sized mills, cane is placed around the cane carrier when it is brought in from the field, Fig. 2. The cane bundles are lifted by horizontal boom derricks onto one or more cane-feeder tables which discharge onto feeder cane carriers, Fig. 3. When there are two derricks, they are placed on each side of the main cane carrier, and the cane-feeder tables also are placed on each side of the main carrier so that they can feed cane from two sides. This arrangement also provides loading from a greater area and stand-by loading facilities if one derrick should break down. Variable control of the feeder tables is necessary to provide the cane carrier constantly with a bed of cane.

In the larger mills, feeder cane carriers arranged either in tandem or at right angles to the main carrier are used in conjunction with feeder tables, truck dumpers, and rail-car dumpers, in various combinations. The feeder cane carrier onto which the

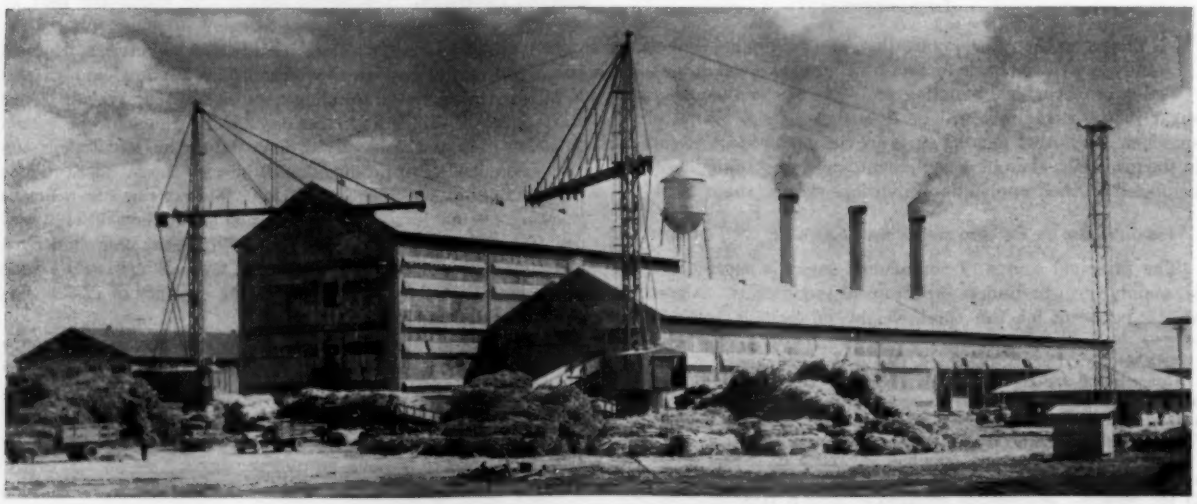


Fig. 2 Cane Bundles Are Placed Around Crane Carriers

cane is discharged has a variable-speed control so that cane can be delivered to the second or main cane carrier uniformly.

Because the cleaned cane comes in varying lengths up to about 10 ft which, when dumped, lie in the cane-carrier trough in all directions, there is considerable air space, so the load is quite deep. This necessitates high sides or skirts made of steel plate, wood, or concrete for the full length of the carrier.

The main cane carrier, which is inclined, has a discharge point about 20 to 30 ft above the ground and just above the receiving hopper or chute to the crushers, but not to exceed 30 deg, and usually is inclined at about 20 deg. Increase over 30 deg decreases efficiency because cane tends to fall back on the conveyer. The main cane carrier is unique in that it is equal in width to the mill rolls. Its length is from 50 to 150 ft, and it has only two other variables, speed and depth of load. These are determined by the capacity of the mill. The average speed of the main cane carriers is 30 fpm. An operator controls the flow of cane to the crushing rolls; so they are subject to starting and stopping fully loaded and require high horsepowers.

Since the regulation of the bagasse blanket is determined at the main cane carrier, there are many factors to be considered. The principal ones are speed and depth of load which were mentioned in the foregoing. To determine its speed, one must consider the amount of cane that the mill tandem can handle, the extraction desired, or the volume of juice that the sugarhouse can handle. While its width is the same as that of the mill rolls, it should be designed for double the capacity not only for possible future expansion, but also it should never be the bottleneck.

A cane leveler or kicker, which consists of several revolving arms, is used on the main carrier before the knife set, or at the head end of feeder carriers to level out the supply of cane to a determined depth, Fig. 4. It is also usual practice to place a knife set which shreds the cane close to the foot or the middle of the inclined main carrier. Prior to the advent of cane knives, which shred the cane for greater juice extraction, wood slats were used on cane conveyers. Because too many cane particles came through the spaces between the wooden slats, which created an unusually dirty condition around the cane carrier, overlapping steel slats have replaced wooden slats in most mills. The slats are mounted on two or three strands of steel roller chain (depending upon the width of the mill). These chains have K-type attachments which have four holes per link to which the slats are attached.

In cane carriers, there is a limitation on the length of centers because of the high proportion of dead weight to live load. Also, the usual operating conditions of 24 hr per day, with abrasion due to dirt and trash from the cane, and corrosion from the juice, plus the load imposed by cane knives, all contribute to give cane carriers a relatively short life expectancy. In sugar mills which have year-round operation, some operators are getting a life of about 3 years. However, where the growing season is around 100 days, the life expectancy of 12 to 15 years is not unusual with lubrication and proper maintenance.

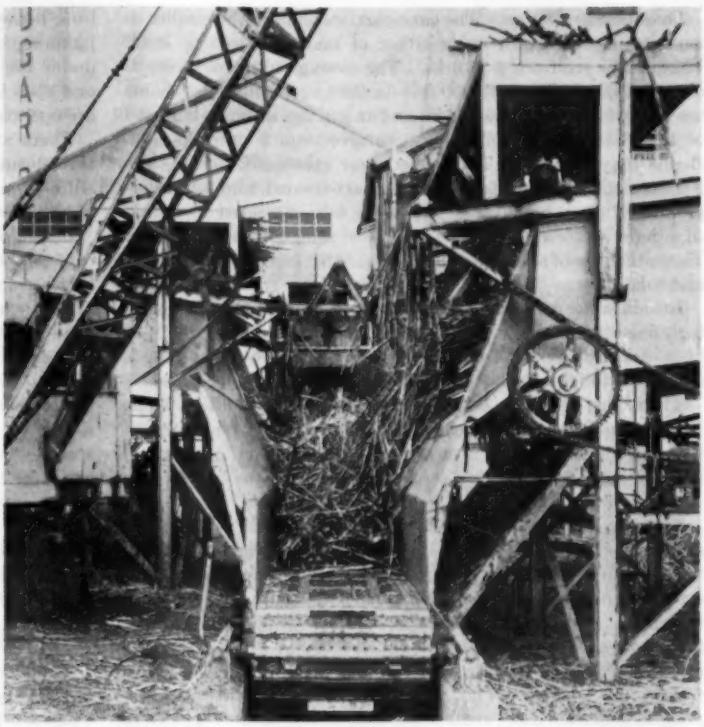


FIG. 3 CRANES LIFT CANE BUNDLES ONTO CANE-FEEDER TABLE

FIG. 4 CANE LEVELER OR KICKER IS USED ON MAIN CARRIER

Cane-Carrier Chains. The cane-carrier chains in the smaller to medium-sized mill are made either of malleable iron or steel. The pitches are from 4 to 6 in. The average ultimate strength range is from 28,000 to 45,000 lb. In the larger mills, steel chains are used with pitches of 6 to 8 in. and an average ultimate strength of 47,000 to 125,000 lb. In the future, even 9 or 12-in-pitch chains may be used. The better steel cane-carrier chains are constructed of high-quality steels, heat-treated and precision-ground to close tolerances. Rollers are heat-treated to obtain superhard surfaces. The pins may be of solid or lubricated type. They are made of alloy steels heat-treated for exceptional strength and toughness.

Bushings are made from alloy steel, hardened and ground to high finish to reduce wear to a minimum. Sidebars are made of high-carbon steel heat-treated to give them high tensile strength and also a tough bearing surface in the pin and bushing holes.

The return run of the chain is either festooned over idler wheels, Fig. 5, which are sometimes rubber-covered to prevent wear on the slats, or where the return run is dragged on rails, small wearing lugs are welded on top of the beads of the slats where they rub on the rails.

Cane-Carrier Slats. Cane-carrier slats are made of wood or steel. Very few mills today have wooden slats because too many trash and cane particles fall through the spaces between the slats, especially on the main cane carrier which has knife sets chopping the cane. The steel slats have formed beads spaced accurately for free flexing with the chain. Their overlap is close, so that it prevents cane from dropping through. They are generally $^{3}/_{16}$ or $^{1}/_{4}$ in. thick. The $^{1}/_{4}$-in. slats are better to resist the pounding of the cane loading as well as the knife set. They usually are bolted to the chains. Sometimes the bolt is put in upside down so that they project above the nut and slot to serve as pushers. Slats do not wear out, but are deformed by tramp iron, heavy loading, the pounding of the knife set, or otherwise jamming. Heavy rails are set under the cane-carrier chains under the knife set to prevent deflection of slats. The chain and slats take a terrific beating at this point because knife sets are operated with as high as 400-hp motors.

There are two and three-beaded slats. The camel back or three-beaded slat is best because the middle bead gives greater stiffness and also holds the cane on the incline better and provides resistance to the cane knives.

Cane-Carrier Sprockets. Because of the heavy loads on cane carriers, steel sprockets should be used; 15-tooth sprockets or larger are recommended for 6-in-pitch chains but never less than 12-tooth or the overlap in slats will open up on going over the head sprockets. For 8-in-pitch chains 13-tooth sprockets are preferred but 11-tooth should be considered minimum.

The drives of cane carriers are by steam engine or electric motor. At the terminals all head-shaft sprockets are keyed firmly to the shaft in line and in sets. On the tail shaft, one sprocket is keyed and set-screwed to the shaft while the others run loose to compensate for uneven wear of the strands of chains.

The latest innovation reported in a Hawaiian sugar mill is the use of closed-circuit television to enable the operator of the cane-carrier system, from a remote point, to see and to regulate the flow of cane to the main cane carrier from the feeder and washer carriers, in order to keep the milling to capacity.

Cane-Feeder Tables

Cane-feeder tables are a part of the cane-carrier system. They are rectangular in shape with the width larger than the length. They are placed alongside the cane carrier to feed cane to it as required and are built with sufficient height to allow a slight downgrade, which makes the cane-moving job easier.

Cane is received in bundles held together by long link-chain

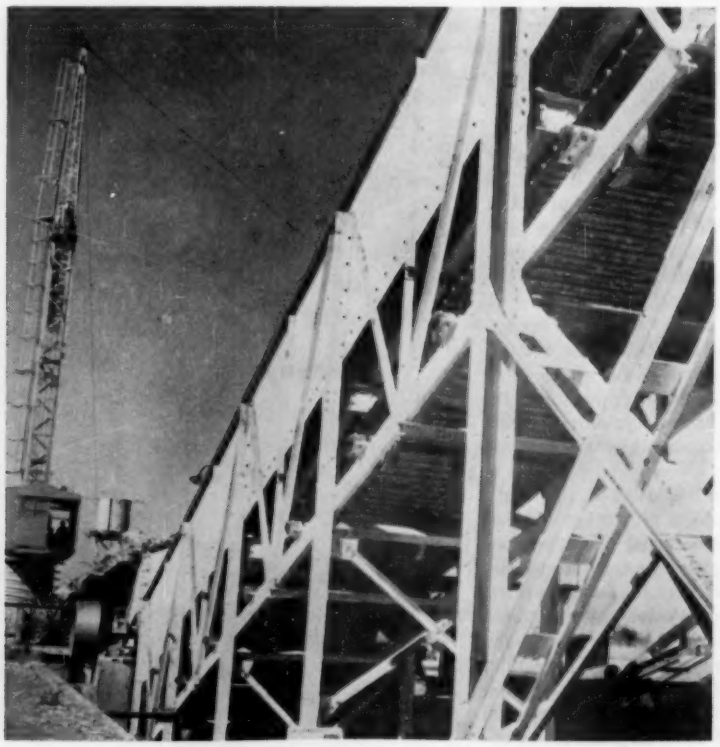


Fig. 5 View of Return Run of Carrier Chain

slings. The hoist picks these up from trucks or piles and holds them over the cane-feeder table. The chain lock is tripped and the cane dumps on the table with considerable impact, Fig. 6.

The average speed of cane-feeder tables is 10 fpm, but a variable speed is needed as feeder tables are not operated at a constant speed. They are started and stopped by an operator in relation to cane requirements on the main carrier. The prime movers are steam engines or electric motors.

Cane-Feeder Chains and Sprockets. The capacity required determines the width, number of chain strands, and centers. Four to twelve strands of chain spaced from 18 in. to 3 ft apart with H-type pusher attachments all operate over head sprockets which are keyed in line on one head shaft. This imposes a tremendous torque on the head shaft. The head shafts are of rather large diameter and usually use several large bearings. It is also advisable to have head sprockets ranging from 24 to 30 in. diam. The tail sprockets are the same size as the head sprockets. However, they are usually keyed to stub foot shafts, which operate in babbitted pillow blocks.

Each chain strand operates in a trough to protect it from the force of the cane dropped on the table. The return run hangs free and its weight serves as an automatic takeup.

Sometimes steel chains are used but malleable-iron and combination chains are in the majority. Corrosion-resistant malleable-type chains which are specially treated and alloyed, made under various names, are advisable because the feeder tables are out in the open without cover, and of course, the sugar-cane juice is corrosive. These special metals also have better physical characteristics than malleable iron which enable them to better resist the shock-loading present in cane-feeder-table operations.

Intermediate Carriers

Intermediate carriers are used to convey the crushed cane from one set of three juice-extraction rolls to the next set. These sets of rolls, or mills, as they often are called, are placed approximately 12 to 15 ft apart. Intermediate carriers are a function of milling rather than an individual conveyer because they operate as a part of the mill train.

Slat conveyers are mainly in use today for this operation. They carry the crushed cane on slats on the top run and return underneath, Fig. 7. They rarely choke the mills and provide a steady bagasse blanket to the succeeding mill. In addition, they also help force the feed.

On intermediate carriers, two or three strands of chains, depending on the width, are used with slats mounted on top. At present, double-sprocket cast-malleable chains with double-beaded overlapping slats are most popular. When new, this overlapping is close and permits very little bagasse to come through and get into the juice. However, when they are deformed because of tramp iron, or are loosened from the chains for other reasons, considerable trash may get through and into the juice. In these cases, occasional hand-cleaning is necessary, or if it is extremely bad, trash drag conveyers are used in the juice pans under the carrier, Fig. 8.

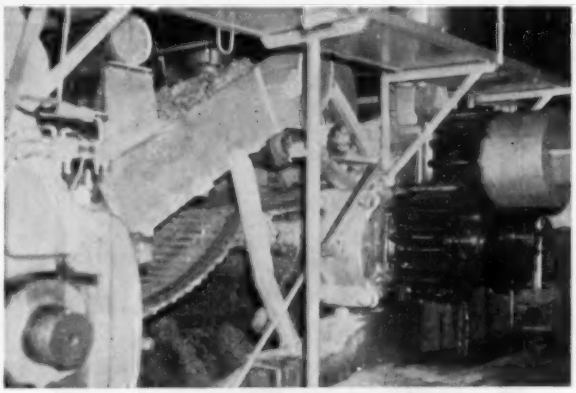


Fig. 7 Intermediate Carriers Employ Slat Conveyers

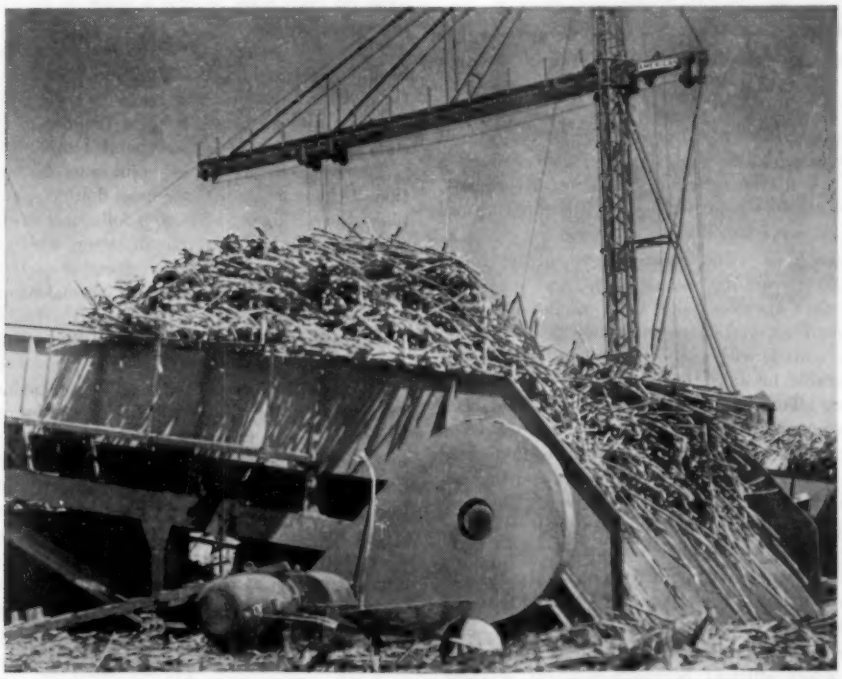


Fig. 6 Cane-Feeder Table

While the width of the intermediate carrier is determined by the width of the mills their speed is the same as the mill-rolls surface speed to slightly faster. If the conveyer speed is less or faster than the mill rolls it is obvious that the bagasse blanket would thin out or thicken. If the bagasse blanket becomes too thick it chokes the mill; if too thin, the full capacity is not attained, and the juice extraction is lower.

The position of the mill rolls is an important factor in the design of the intermediate carrier. The mill rolls are in a triangular position with two rolls at the bottom and one roll on top which exerts the crushing pressure. The intermediate carrier must take on a shape so that the tail end is in a low position to receive the bagasse blanket from the bottom discharge roll of the previous mill, and the head end high to a drive shaft with sprockets, and from there forward and down to a nose shaft, to bring the bagasse into the space between the bottom receiving roll and the top roll of the subsequent mill. On the return run, the conveyer is usually carried over idler wheels, or allowed to slide on steel ways. At the tail end, it usually operates over idler rolls or sprockets.

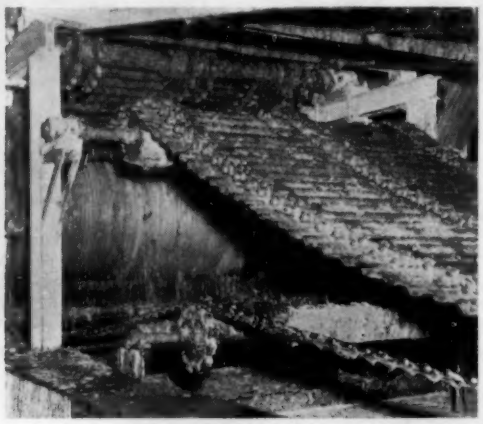


FIG. 8 DOUBLE-SPROCKET STEEL CHAINS WITH DOUBLE-BEADED OVERLAPPING SLATS

Since the intermediate carrier is driven as a part of the mill train, its power is taken off the mill-roll drive, then through a clutch and chain drive to the intermediate-carrier drive shaft. The frame of the intermediate carrier is integral with the mill but is demountable for servicing, Fig. 9.

Intermediate-Carrier Slats. The main components of intermediate apron conveyers are the chains, slats, and sprockets. Single-beaded steel slats are used with short pitched chains in small mills. The medium and large mills use double-beaded slats. These slats are made with shallow beads and deep beads. The latter are preferable because they provide greater stiffness as well as better feed. To minimize deforming due to load, slats are supported by two or three strands of chains which are set on about 26-in. centers. The better-made slats have die-formed beads for accuracy of overlap, and to provide a flat portion between the beads to seat squarely the bolt or rivet. This prevents the corrosive juices from getting under the bolt or rivet head and permits the slat to seat well on the chain attachment.

Slats are held to the chain attachment with one $^5/_8$-in-diam or two $^3/_8$-in-diam rivets or bolts, Fig. 10. It is believed that one $^5/_8$-in-diam bolt or rivet is better than two $^3/_8$-in. size.

Care must be exercised, when using bolts and nuts to assemble slats to chain, to select efficient corrosion-resistant washers. Nevertheless, occasional check-up is important because lock washers may break, bolts may crystallize, that is, crack because of corrosion fatigue, and nuts may back off.

The direction of travel of slats is controversial. Some engineers contend that the overlapping bead should be forward, so that the edge of the slat will act as a pusher and feed the mills. Others contend that if they are operated in this direction, bagasse is forced into the space between the beads which distort and loosens the slats from the chains, thus causing more space for trash to be dropped into the juice pans.

The author prefers the use of deep-beaded slats to be operated with the larger bead trailing, Fig. 11. This places the opening between slats on the rear. The larger or overlapping bead helps feed the mill by the knuckling action as it goes over the head and nose sprocket. Since the forward bead is under the larger overlapping bead of the previous slat, a negligible amount of trash can get into the space between the beads.

Slats are available in 10 gage, $^3/_{16}$ in., and in some styles $^1/_4$ in. thick. Because of corrosion by sugar juices, and to resist distortion, it is felt that the $^3/_{16}$-in-thick double-beaded slats should be adequate. Slats do not wear out but are bent out of shape because of tramp iron or other damage before they corrode out.

Intermediate-Carrier Chains. Intermediate-carrier chains for slat-type conveyers are of single or double-sprocket types. The single-sprocket type has either K or E-attachments. On this type chain, bagasse, which comes through the space between the slats, packs in the pocket between the side bars of the chain and the slat to which it is attached. This causes the chain to ride higher and higher on the sprocket teeth and eventually jump off. Also, during this process of packing, the load on the chain increases and it stretches out of pitch. This causes faster wear on the sprockets, chain, and so on, but worst of all is the fact that it jams and causes much down time to remedy the situation. The double-sprocket-type chain has become popular because this problem is eliminated. This type of chain has round barrels or trunnions on either side cast integral with the side bar, Figs. 10 and 11. The double-sprocket teeth thus engage the teeth on both sides leaving a space in between for trash to fall through, Fig. 12.

These double-sprocket chains are available with easily replaceable pins and bushings. In addition, they have a large area both in the pad, at the narrow end, and under the side bars, on which they slide, as well as large-diameter trunnions, for long wear. The slats are attached to an E-type attachment.

There is a divergence of opinion as to the direction of operation of these chains. Since most double-sprocket chains have replaceable bushings and pins, it follows that they should be operated closed end forward like an arrow, which places the wear between the pins and bushings instead of on the trunnions, Fig. 11.

Intermediate chains made of special-process alloy malleable iron are well worth the extra cost. Used with noncorrosive pins and bushings, they make an excellent combination to resist the corrosion by the cane juice.

Intermediate-Carrier Sprockets. Intermediate-carrier sprockets should have not less than 10 teeth or the slat overlap will open and let trash through when operating over the sprocket. They should be as large as possible space will allow. They should be keyed in line and in sets. It also is recommended that chilled-tooth cast-iron sprockets cast integral for tooth alignment be used.

The modern trend is for two-shaft intermediate carriers, that is, a head shaft and a tail shaft eliminating the nose shaft. As large a sprocket as 24 teeth may be used on the head shaft and is set so that the conveyer blanket is fed to the next mills at the proper place for grinding.

JUICE STRAINERS

The juice, as extracted from the mills, contains some refuse

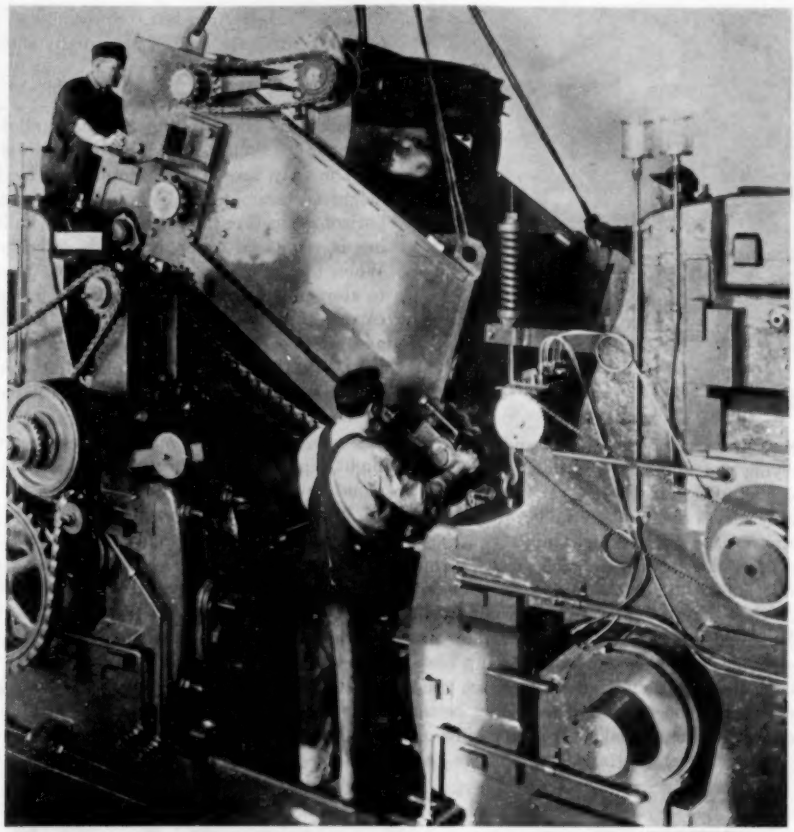
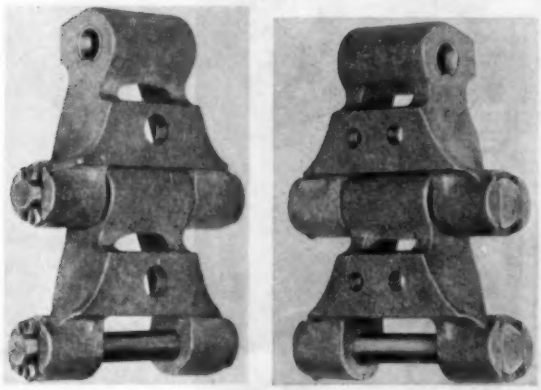


FIG. 9 FRAME OF INTERMEDIATE CARRIER IS INTEGRAL WITH MILL



Single-attachment hole Double-attachment hole

FIG. 10 DOUBLE-SPROCKET-TYPE INTERMEDIATE-CARRIER CHAINS

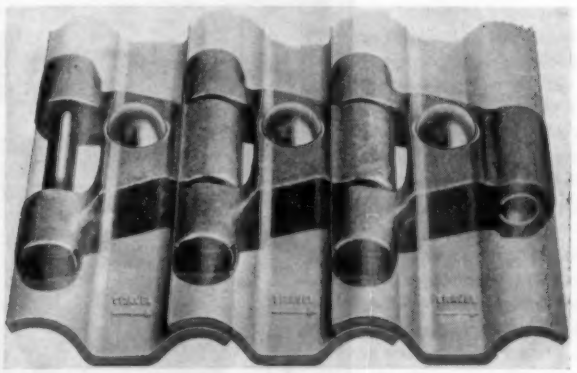
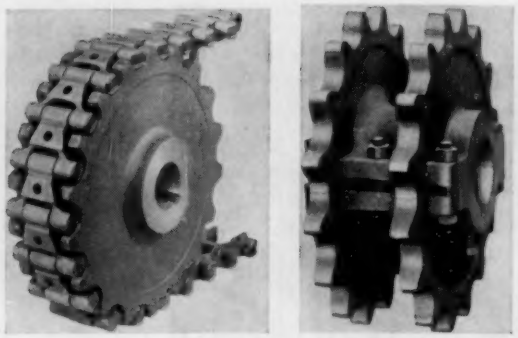


FIG. 11 DEEP-BEADED SLATS

consisting of small pieces of cane fiber, leaves, and other trash. Juice strainers, cush cush, or trash elevators as they are sometimes called, separate this trash from the juice, elevating it to a screw conveyer which returns it to the stream of crushed cane on the intermediate carriers, Fig. 13. Juice strainers also are shown on the right side of the tandem, Fig. 1.

The juice and strainer conveyer itself consists of two strands of chain with F-type attachments spaced about every 8 to 10 chain pitches to which are attached wood or steel flights having rubber scraper strips, and sometimes brushes on the bottom edge. (Very small mills may use a one-strand conveyer.) These flights operate in a trough into which the cane juice pours over a perforated copper plate which extends the length of the bottom horizontal section as well as a portion of the inclined section of the trough. Under the perforated copper plate there is usually stiff woven-wire screen or other devices to prevent sagging so that the scraper flight bottom can ride flush with the perforated plate and give a squeegee action.

The refuse is dragged up the incline and discharged into a screw conveyer which in turn returns the refuse to the inter-



Solid Split

FIG. 12 DOUBLE-SPROCKET-TYPE CHAIN DRIVE

mediate carrier. The return run of the chains and flights is located above the carrying run and is either dragged back on steel ways or festooned over idler wheels.

A juice strainer usually serves one to three mills, so there are usually one or two of appropriate size in a sugar mill.

Juice-Strainer Chains. Juice-strainer chains usually are made of malleable iron; 2.609 and 3.075-in-pitch cast chains are used most widely. Cast detachable and pintle chains with F-attachments are illustrated in Fig. 13. Because they are constantly bathed in cane juice which is corrosive, special-process alloyed malleables, made under various trade names, are used widely. These not only resist the corrosion, but have better physical characteristics than ordinary malleable iron. Links of this metal used with stainless-steel pins and brass cotters give excellent service.

Juice-Strainer Sprockets. Sprockets of cast iron with chilled teeth are adequate. They are usually about 24 in. pitch diam.

BAGASSE CARRIERS

Bagasse is chuted from the last mill to the bagasse carrier after the maximum amount of juice is extracted. Bagasse carriers usually are in two sections. The inclined section gives height necessary to feed the boilers by gravity, Fig. 14, and the horizontal section provides feed to the boilers, Fig. 15; this feeding is regulated through double-acting rack-and-pinion gates. When the volume of bagasse is more than necessary, it is sent to storage and later recovered from storage. When the bagasse carrier that does this function of feeding, bringing to storage, and recovering has the form of a loop, it is called a bagasse loop, Fig. 16. If boilers are placed at right angles to the line of the main bagasse carrier, lateral bagasse carriers receive a portion of the discharge from the main carrier and feed these boilers.

Bagasse-Carrier Chains. Bagasse carriers are of scraper or flight type, operating in a trough. Two strands of chain, having wood or steel flights attached, are used. The flights are set 18 to 30 in. apart. Depending on size of the conveyer, 2.608, 3.075, 4, and 6-in-pitch chains are used. Bagasse carriers are designed to travel at 60 to 125 fpm, depending on the capacity required, width and depth of load carried by each flight. Like any conveyer the slower they can be run, the longer the life.

Sometimes, very fine bagasse called bagacillo is collected under the inclined or horizontal section. A screen is placed at the bottom of the trough to collect the fine bagasse which is used as a filter medium in some makes of juice filters. A screw conveyer, small scraper conveyer, or pneumatic conveyer is used to transport this fine bagasse from the conveyers to the filters which are located in the sugarhouse.

FIG. 13 JUICE STRAINER OR TRASH ELEVATOR

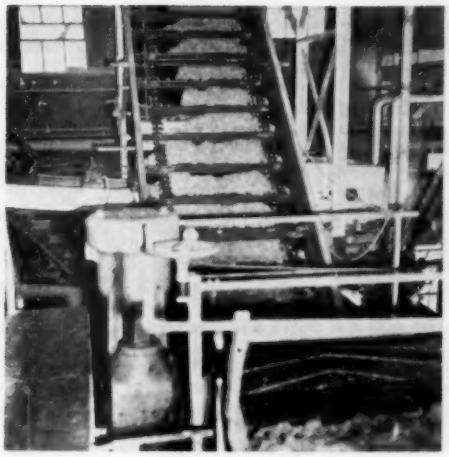


FIG. 14 INCLINED SECTION OF BAGASSE CARRIER

FIG. 15 HORIZONTAL SECTION OF BAGASSE CARRIER

The smaller-pitch chains are used in the smaller mills and are usually without rollers. However, the medium and larger mills use 4 to 6-in-pitch chains having 2 to 3-in-diam rollers to provide rolling instead of sliding friction. These permit longer centered conveyers. Large rollers are recommended because it means there will be less wear in the live bearing of the chain since they revolve less per linear foot of conveyer than smaller-diameter rollers. The bagasse is so light to handle (approximately 7 to 11 lb per cu ft) that often the weight of the chains and slats per foot of conveyer is greater than the material handled. Chains with heat-treated side bars which provide higher tensile strength are used in the long bagasse loops of several hundred feet. In these long conveyers steel chains are recommended.

The medium-sized mills sometimes use cast-malleable-iron roller chains whose average ultimate tensile strength runs 12,000 to 34,000 lb and pitch from 4 to 6 in. But in the larger mills 6-in-pitch steel chains with strengths ranging from 38,000 to 55,000 lb average ultimate strength are used.

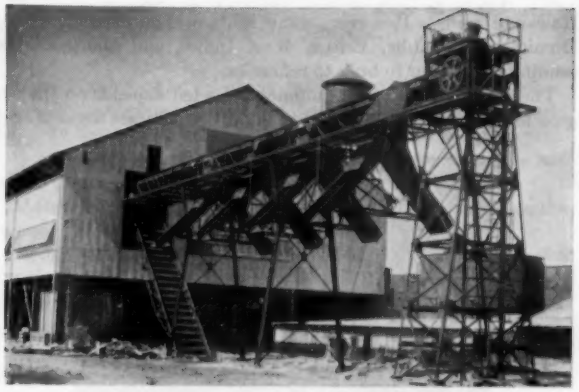


FIG. 16 BAGASSE LOOP

Bagasse-Carrier Sprockets. Chilled-tooth cast-iron or cast-steel sprockets are adequate for bagasse-carrier service. As large a sprocket as space permits should be used. However, for 4-in-pitch chains about 18-tooth sprockets and for 6-in-pitch chains about 15-tooth sprockets or larger should be used, but they should not be less than 12 teeth in any case.

MISCELLANEOUS CONVEYERS

In addition to the principal types of conveyers in the sugar mill, many other miscellaneous types are used in the sugar refinery. While brief mention will be made there will be no attempt to treat each of these in detail.

Screw conveyers of the helicoid-ribbon type handle sticky materials such as massecuite and collect raw sugar from the centrifugals. Helicoid solid type handle bagacillo and muds as well as raw sugar. Sometimes oscillating or grasshopper-type conveyers are used for raw sugar after it is dropped from the centrifugals. This in turn discharges into the boot of the bucket elevator. The sugar elevator raises the sugar to a hopper from which sugar is bagged and weighed. Bucket elevators for handling raw and moist refined sugar are equipped with style C buckets, which are in the form of a right angle; style A buckets are used for free-flowing or dry sugar. In a few mills en masse conveyers which consist of solid flights attached at intervals to endless chain or wire ropes operating in a casing are used for handling free-flowing sugar.

Almost every sugar mill has portable conveyers, Fig. 17, or elevators for handling bags from the scales, bag-stacker conveyers for piling bags in the warehouse, or for handling bags between the warehouse and railroad cars. Fixed or stationary platform-type conveyers are used in main carriers in long warehouses, or along a railroad siding for bag handling. Belt conveyers also are used for bag-handling or handling in bulk, as shown in Fig. 18.

Most mills do not have their own refineries; therefore, a few years ago, practically all raw sugar was delivered to refineries in burlap bags of 200 to 325 lb, and much still is. However, handling, raw or refined sugar in bulk obviously saves time and also the cost of the burlap bags, of sacking and unsacking, of rehandling and cargo space. During the war the burlap-bag supply coming from the Orient was shut off so that while some mills substituted paper bags, others developed bulk-handling systems which fit their particular needs.

While bulk-handling systems vary, belt conveyers combined with bucket elevators are used in various combinations in most mills. The largest percentage of bulk-sugar handling is done by

Hawaiian mills. However, some mills in Puerto Rico, Cuba, Dominican Republic, British West Indies, and Louisiana are sending their sugar in bulk to refineries.

The type of conveying equipment selected depends on the degree to which sugar is handled in bulk. Sometimes sugar is sent

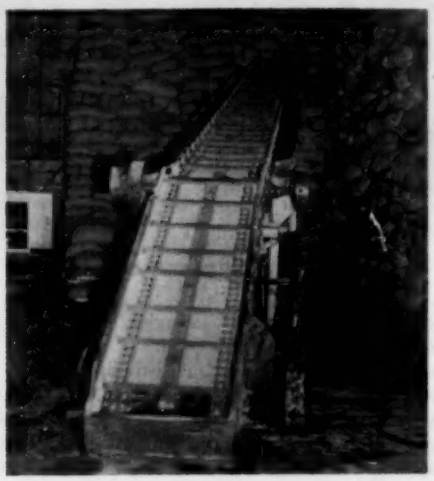


FIG. 17 PORTABLE ELEVATORS FIND WIDE USE

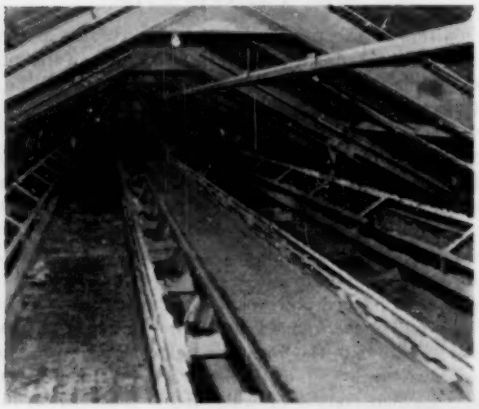


FIG. 18 BELT CONVEYER FOR BAG-HANDLING OR BULK MOVEMENT

to the port in bags to be emptied into ships' holds and then the empty bags are returned to the mills to be refilled. The distance of the mill from the loading docks and the manner of transport of bulk sugar to docks, as well as handling facilities at the mills and the dock, are very important. If the mill is close to the dock, belt conveying is used directly to the ship's holds. If the mill is far from the docks, then easily unloaded tote boxes or bins may be used for transportation to dockside, where they are hoisted and dumped into the ships' holds, and then returned to the mill to be refilled.

At the mill, as raw sugar is made, it is handled by belt conveyer to storage or large silos or bins; bucket elevators may be used in conjunction with belt conveyers for filling the bins. Clamshell buckets and belt conveyers usually reclaim sugar from storage for sending to the port.

Sugar ships are being designed like ore boats for fast unloading of bulk sugar. In some cases marine legs, which are simply bucket elevators which can be moved downward into the hold to unload sugar, are used in conjunction with belt conveyers to send the bulk sugar along to the refinery. In the refinery various combinations of clamshell buckets, bucket elevators, and belt conveyers handle the raw and refined sugar in and out of storage.

CONCLUSION

To summarize briefly, therefore, the main types of conveyers used in the cane-sugar-making process are slat or apron conveyers in cane and intermediate carriers, flight or scraper conveyers in juice strainers, bagasse carriers, and portable bag stackers; screw conveyers and grasshopper conveyers in sugar handling from the centrifugals, platform conveyers, bucket elevators, and belt conveyers in raw and refined-sugar handling.

ACKNOWLEDGMENT

Acknowledgment is hereby given to members of the Conveyer and the Chain Engineering Departments and others at Chain Belt Company for help and criticism of this presentation.

Discussion

P. J. IMSE.[1] This paper gives an excellent presentation of the basic layout, function, and value of conveyers in cane-sugar mills. All phases of such a subject cannot be covered in a single paper. We, in the chain-manufacturing industry, are continuously interested in the development of chains used in the sugar industry.

New demands are put on conveyer chains because of constantly increasing mill capacity. New materials are becoming economically feasible for use in chain manufacture. Newly developed manufacturing techniques make new designs practical. All of these factors make conveyer-chain development an item of constant interest.

One of the best sources of information available to a chain manufacturer to guide him in the logical improvement of his product is a sample of a chain that failed to do the expected job, together with a complete story of the conditions surrounding the failure, and a complete set of operating data. Mill operators can do much to help in the development of conveyer chains that will give them greater service if they make the effort to save such samples, compile the data, and place them in the hands of a representative of a chain manufacturer.

It should be emphasized that conveyer-chain maintenance is very economical. For example, it may cost 50 to 100 per cent more money to buy a chain that will give double the life on a particular application, but the operator of a mill can cut his conveyer service life by five or more times, merely by inadequate maintenance. Proper "between-season" care, removal of interferences, proper slack adjustment, sprocket alignment, and so on, can do more to increase chain life than an expensive increase in the chain design.

R. C. SOLLENBERGER.[2] The author has done such an excellent job of presenting the basic facts about machinery operating under the conditions found in cane-sugar mills that it is hard to do anything other than to emphasize certain of his points.

The writer's experience in a variety of industries leads him to comment that operating conditions in a sugar mill are about as severe as those found in any industry. This, combined with the usual short grinding season, poses some problems that are unique. It is difficult to think of an industry where "preventive maintenance" can be applied with more profitable results.

Experienced design engineering is most essential to nonstop operation of the mill during the grind. There are excellent reasons

[1] Chief Engineer, Chain and Transmission Division, Chain Belt Company, Milwaukee, Wis. Mem. ASME.

[2] Executive Vice-President, Conveyor Equipment Manufacturers Association, Washington, D. C. Mem. ASME.

why the features mentioned by the author in connection with carrier chain are used. This is an example of design evolution dictated by the specialized needs of the sugar mill.

Another example is the trend away from sleeve bearings to ball or roller bearings. The combination of slightly acidic juice and abrasive dust is soon fatal to unprotected bearing surfaces. In the writer's opinion, ball and roller bearings should be fitted with seals of considerably better than normal efficacy if satisfactory operation is to be obtained. The writer does not know of any application where the polished surfaces of races and rollers or balls will be ruined more quickly than around the cane carriers and knives if juice and dust can get to them.

Ordinary felt seals in contact with the shaft or bearing mounting sleeve offer little protection. The writer is a firm believer in heavy-gage metal labyrinth seals, the outer member of which will act through centrifugal force to direct foreign material outward and away from the bearing.

The comments on protection of bearings apply with equal force to other operating parts such as friction clutches, speed reducers, chain drives, and even belt drives.

The best-designed machinery in the world won't give satisfactory performance or trouble-free operation unless it is backed up by a regular and systematic maintenance schedule which provides for adequate but not overlubrication. The latter is especially important where high-speed antifriction bearings are concerned.

It is not possible to beat the combination of sound design backed up by a good maintenance program.

J. B. Webb.[4] The following brief comments are offered:

Costly shutdowns may be cut down by maintaining spare parts, stand-by drives, and using limit switches instead of shear pins.

It is suggested that provision be made for cleaning slats and chain.

Automatic lubrication would aid in prolonging life of equipment.

Keystone chain has been used successfully during the past 5 years or more in the logging industry. This type might be considered for applications described in the paper.

Fire prevention should be given some thought.

Belt conveyers would be applicable except on extremely wide feeders.

Variable-speed drives might be considered in this instance.

Regarding direction of travel of slats, we believe that the overlapping bead should be forward.

Fig. 12 is a good idea.

What percentage of waste is involved in handling methods described?

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[4] President and General Manager, J. B. Webb Company, Detroit, Mich. Mem. ASME.

Contribution to the Knowledge of Pressure Measurements During Metal Deformation

By J. FRISCH,[1] BERKELEY, CALIF.

Two pressure-measuring devices using SR-4 strain gages and steel pins of 1/8 in. and 1/4 in. diam, respectively, were inserted in the wall of an extrusion cylinder. During the extrusion of commercially pure lead the mean pressure and radial wall stresses were measured and found to be in close agreement with the calculated values. The calibration of the instruments showed considerable difference between the curves obtained from dead-weight calibration and those found from a simulation of the extrusion process.

Introduction

DURING the deformation of metals by forming and shaping processes such as forging, rolling, extruding, and metal cutting, the pressure and stress distribution within the material may be difficult to determine since a knowledge of the geometry of the specimen and the load distribution may be insufficient. Although Pearson[2] published a considerable amount of data on the pressure of extrusion, an experimental investigation of the pressure along the cylinder wall would facilitate greatly the study of stress distribution within the metal. Such a study for a billet undergoing plastic deformation during an indirect extrusion process was suggested by Thomsen and Lapsley.[3] In their articles on the mechanics of plastic deformation of metals Thomsen, et al.,[4] pointed out that while a knowledge of the applied load may be sufficient to determine the stress pattern, in the case of nonuniform loading over the cross section of the billet the experimentally obtained mean pressure, even at one point, would make it possible to apply the equations of plasticity although the load at any given point is unknown.

The use of a pressure gage for finding the pressures in rolls during calendering and rolling processes was demonstrated successfully by Bergen and Scott[5] as well as by Lueg.[6] Since the determination of the mean pressure on the inside surface of the cylinder during extrusion presented essentially the same problem, a pressure-measuring device employing a steel pin and bonded-wire strain gages was inserted in the wall of the extrusion cylinder. A 1/8-in. steel pin was used but the question of the size effect led to the inclusion of a second gage with a 1/4-in. pin inserted in the cylinder at the same elevation, as shown in Fig. 1. The area ratio of 4 to 1 between the pins was found to be large enough to demonstrate that the smaller pins will yield more accurate results in relation to the mean pressure rather than the average pressure, i.e., the applied load divided by the billet area. In addition to the mean pressure p and the average pressure p_{avg}, the wall pressure p_w, which is the stress per unit area experienced by the gage pins and approximately by the cylinder wall, is of general interest since the design of the extrusion chamber should be based on the wall pressure. The distinction between these three pressures necessitated the careful calibration of the instrument, the gage response under dead-weight calibration being different from that under a simulation of the indirect extrusion process used in the investigation.

Experimental Apparatus and Procedure

The equipment used for the determination of pressures and stresses during the indirect extrusion of commercially pure lead at room temperature is shown in Fig. 1. The extrusion cylinder which has an inside diameter of 4.3 in. is equipped with two pressure gages using 1/8-in. and 1/4-in-diam steel pins inserted in the cylinder wall but otherwise of identical design. Each pin is in contact with a round bar which has been ground flat to accommodate two bonded resistance-wire strain gages (Baldwin SR-4 gages, Type A-7). The gages are connected in a parallel

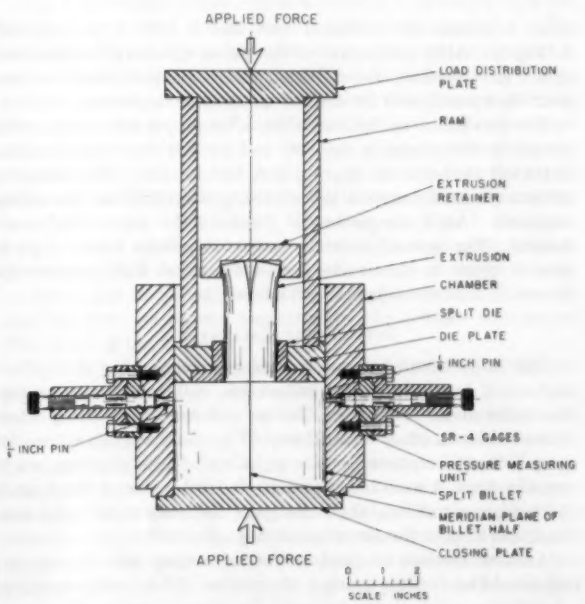

Fig. 1 Sectional View of Indirect Extrusion Apparatus and SR-4 Pressure

[1] Assistant Professor, Engineering Design Division, University of California. Assoc. Mem. ASME.

[2] "The Extrusion of Metals," by C. E. Pearson, John Wiley & Sons, Inc., New York, N. Y., 1944.

[3] "Experimental Stress Determination Within a Metal," by E. G. Thomsen and J. T. Lapsley, Jr., Proceedings of the Society for Experimental Stress Analysis, vol. 11, no. 2, 1952, pp. 59–68.

[4] "An Experimental Investigation of the Mechanics of Plastic Deformation of Metals," by E. G. Thomsen, C. T. Yang, and J. B. Bierbower (submitted for publication in the *Engineering Series of the University of California*).

[5] "Pressure Distribution in the Calendering of Plastic Materials," by J. T. Bergen and G. W. Scott, Jr., *Journal of Applied Mechanics*, Trans. ASME, vol. 73, 1951, pp. 101–106.

[6] "Spannungsverteilung und Werkstoffluss im Walzspalt," by W. Lueg, *Stahl und Eisen*, vol. 14, 1933, pp. 346–352.

Contributed by the Research Committee on Metal Processing and presented at a joint session of the Mexico Section and Research Committee on Metal Processing at the International Meeting, Mexico City, Mexico, March 10–12, 1954, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received by ASME Headquarters, February 1, 1954. Paper No. 54—Mex-19.

electric bridge circuit to a standard SR-4 recorder such that the bridge output was only a function of the axial strains in the pins. An adjusting screw was provided for alignment of the end surface of the gage pin with the cylinder wall. The sensitivity of the gages is 30 psi per microinch of reading. The indicator having a least count of 10 microinches can be estimated to 5 microinches or better. In view of the fact that the pressures encountered during the extrusion of lead fall between 6000 psi and 11,000 psi, the accuracy of the instrumentation was only in error from $1^1/_2$ to 3 per cent.

The die plate used for the extrusion of a solid bar had a sharp-edged hole with a diameter of 1.5 in. so that the area ratio of the cross sections of the billet to the extruded bar is 8.2 to 1. For the calibration of the gages under no-flow conditions a solid plate was used to replace the die. All tests were performed in a 200,000-lb tensile testing machine. While the axial loads were recorded by the machine, a dial indicator of 0.001 in. accuracy and a stop watch were employed to find the die speed, and a plot of die movement as a function of extrusion time is shown in Fig. 2. The area ratio of 8.2 to 1, which is also the ratio of the extrusion rate to die rate, was applied to the values of die move-

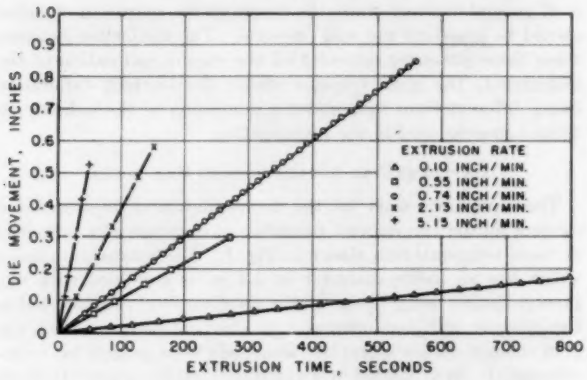

Fig. 2 Die Movement as Function of Extrusion Time

ment to obtain the extrusion rates of 0.1, 0.55, 0.74, 2.13, and 5.15 ipm. After calibration of the gages, which will be discussed in the next section, the extrusion tests were performed for constant-flow conditions for the extrusion rates mentioned.

The two halves of the lead billet, after proper lubrication, were placed in the extrusion chamber and a 1.5-in-diam solid bar was extruded through the sharp-edged circular die. The extrusion rates were held constant by adjusting the valves of the testing machine. After completion of the tests the gages were recalibrated. For several tests the extrusion process was a stepwise one in order to determine the experimental flow patterns discussed in a previous paper by Thomsen, et al.[4]

Instrument Calibration

The calibration of the pressure gages consisted of three different steps, i.e., dead-weight calibration, calibration using a rubber billet under no-flow conditions, and calibration using a lead billet under no-flow conditions. The dead-weight calibration gave a linear response for the $^1/_8$-in. and $^1/_4$-in. gages as can be seen in Figs. 3 and 4, respectively. The slopes of the lines in both figures indicate that the gage response under pure axial load was almost the same for both gages.

Another method of dead-weight calibration was the use of a rubber billet. After placing the rubber billet in the extrusion chamber a solid plate was used for closure and application of compressive loads. The load divided by the billet area, when

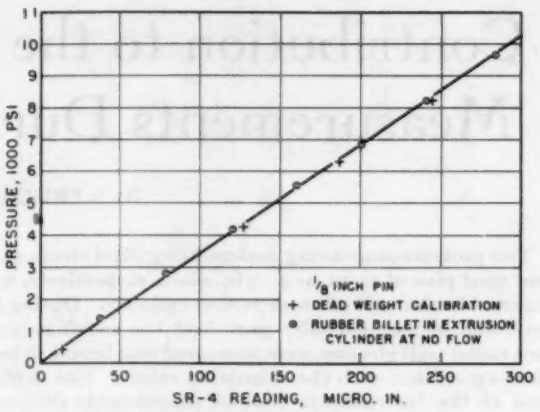
PRESSURE, 1000 PSI
SR-4 READING, MICRO. IN.
1/8 INCH PIN
DEAD WEIGHT CALIBRATION
RUBBER BILLET IN EXTRUSION CYLINDER AT NO FLOW

Fig. 3 Dead-Weight and Rubber-Billet Calibration Curve for $^1/_8$-In. Gage; No-Flow Condition

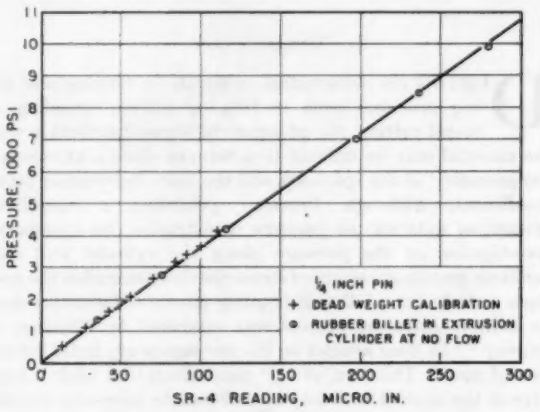
PRESSURE, 1000 PSI
SR-4 READING, MICRO. IN.
1/4 INCH PIN
DEAD WEIGHT CALIBRATION
RUBBER BILLET IN EXTRUSION CYLINDER AT NO FLOW

Fig. 4 Dead-Weight and Rubber-Billet Calibration Curve for $^1/_4$-In. Gage; No-Flow Condition

plotted against the SR-4 readings, yielded calibration curves for the $^1/_8$-in. and $^1/_4$-in. gages which were the same as those obtained from dead weights. Figs. 3 and 4 therefore show the experimental points of both types of tests and because of the easier method involved, the rubber-billet calibration was the preferred one.

During extrusion for various rates, the wall pressures were determined using the calibration curves of Figs. 3 and 4 and are shown in Fig. 5. The almost linear increase of applied load and corresponding linear increase of average pressure with increasing extrusion rate is also shown in that graph. The measured wall pressure, as recorded by both gages, is seen to be consistently less than the average pressure. This fact is of significance in the design of extrusion cylinders because the average pressure could be used safely in the determination of the maximum stresses in the cylinder. Since both the average pressure and the wall pressure were considerably below the mean pressure, as predicted from the stress analysis for a point corresponding to the position of the gage-pin center lines, a third calibration test using a lead billet was performed. The lead billet was subjected to compressive loads under no-flow conditions in the same manner as the rubber billet.

The calibration curves for the $^1/_8$-in. and $^1/_4$-in. gages are shown in Figs. 6 and 7, respectively. The straight line in each figure represents the dead-weight calibration. Curves *a*, *b*, *c*, and *d* are the mean-pressure variations during loading and unloading as recorded by the strain indicator. The ordinate represents the

mean pressure p, which is constant throughout the billet for any given load. From curve a, it can be seen that the gage does not respond immediately but that a considerable load must be applied before the curve becomes parallel to the dead-weight calibration curve. Upon unloading, the pressure decreases along curve b but the strain-indicator readings do not return to the initial zero reading. Reloading of the billet produces a pressure increase as shown by curve c and subsequent unloading causes a pressure decrease along d. Continued loading and unloading again produces pressures along curves c and d.

In a previous paper,[7] it was shown that the pressure gages would not respond until the metal in the cylinder reached a critical stress. After the billet expanded to the diameter of the cylinder it must be assumed that a small lead plug is extruded into the bore of the receding gage pin. Upon unloading, the contracting cylinder will cause a decrease in the diameter of the lead plug which keeps the gage pin from returning to its initial position.

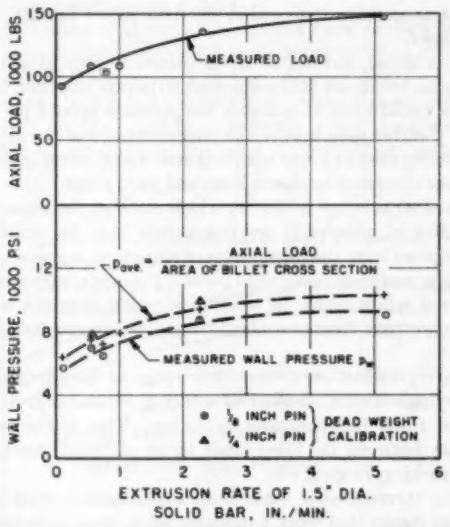
AXIAL LOAD, 1000 LBS
MEASURED LOAD
WALL PRESSURE, 1000 PSI
AXIAL LOAD / AREA OF BILLET CROSS SECTION
MEASURED WALL PRESSURE
1/8 INCH PIN
1/4 INCH PIN
DEAD WEIGHT CALIBRATION
EXTRUSION RATE OF 1.5" DIA. SOLID BAR, IN./MIN.

Fig. 5 Axial Load, Average Pressure, and Wall Pressure as Functions of Extrusion Rate
(Distance between pressure gages and die is 0.4 in.)

Curve c in Figs. 6 and 7 shows a smaller change in strain response for reloading of the billet than curve a representing the initial loading condition. The fact that the existing lead plug is not only being pushed into the gage-pin holes but also experiences an increase in diameter, must account for the smaller axial strain on the pin, and therefore a response of the strain indicator as shown by curves c and d. This calibration, which can be repeated without change, substantiates the usability of the curves for finding the mean pressures experimentally.

During continuous extrusion of lead the gage response is essentially the same as for the initial run on the lead-calibration curves and therefore curve a in Figs. 6 and 7 would appear to be the logical one for finding the mean pressure. However, should the load be lowered during the extrusion, curve c would have to be used. After careful recalibration it was found that the lead-calibration curve for the 1/8-in. gage was somewhat more accurate than the preliminary one shown in the paper by Frisch and Thomsen.[7]

[7] "An Experimental Study of Metal Extrusions at Various Strain Rates," by J. Frisch and E. G. Thomsen, Trans. ASME, vol. 76, 1954, pp. 599–606.

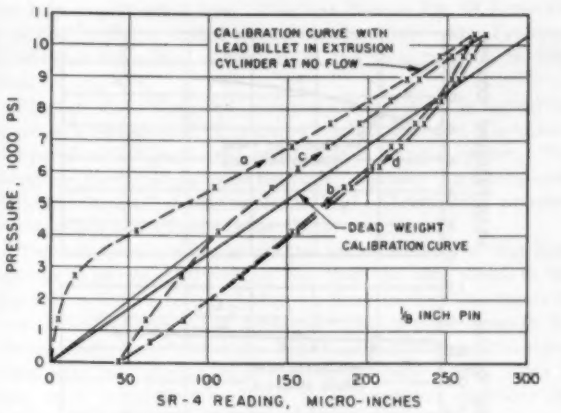
CALIBRATION CURVE WITH LEAD BILLET IN EXTRUSION CYLINDER AT NO FLOW
DEAD WEIGHT CALIBRATION CURVE
1/8 INCH PIN
PRESSURE, 1000 PSI
SR-4 READING, MICRO-INCHES

Fig. 6 Mean-Pressure Calibration Curve for 1/8-In. Gage

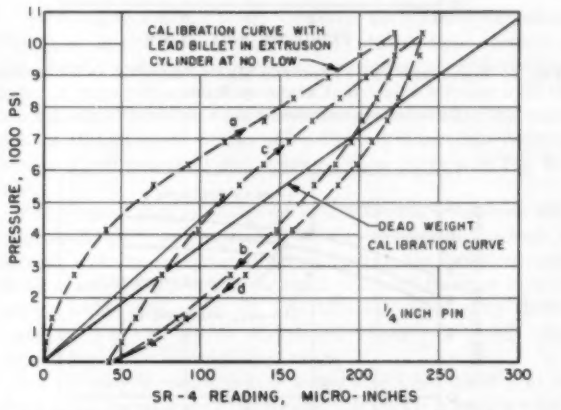
CALIBRATION CURVE WITH LEAD BILLET IN EXTRUSION CYLINDER AT NO FLOW
DEAD WEIGHT CALIBRATION CURVE
1/4 INCH PIN
PRESSURE, 1000 PSI
SR-4 READING, MICRO-INCHES

Fig. 7 Mean-Pressure Calibration Curve for 1/4-In. Gage

Results and Discussion

Fig. 5 shows the applied load as a function of extrusion rate and is found to increase almost linearly with increasing rates. The corresponding average pressure therefore shows the same characteristics but the wall pressure, while also increasing linearly with extrusion speed, is consistently lower than the average pressure. The mean pressure found experimentally is in close agreement with the predicted values and the plot of mean pressure versus extrusion rate is shown in Fig. 8. It can be seen that the 1/8-in. gage records values closer to the actual mean pressure than those recorded by the 1/4-in. gage. However, in Fig. 5 the pressures recorded by the 1/8-in. gage more closely agree with average pressures. Until further tests are performed the conclusion must be drawn that the larger pin shows a size effect in that it records only the same pressures experienced by the cylinder wall rather than those of the metal in the billet.

Fig. 9 is a graph of the three pressures as functions of the five extrusion rates used in the investigation. It is of interest to note that the wall pressure and the mean pressure deviate from the average pressure, i.e., the load divided by the billet area, by approximately the same-percentage error. The wall pressure, as determined from dead-weight calibration curves, is considerably below the mean pressure so that, while the values thus obtained are sufficiently accurate for the design of extrusion equipment, the gage readings based on the lead-calibration curves are the ones to be used for a stress analysis of plastic flow during extrusion.

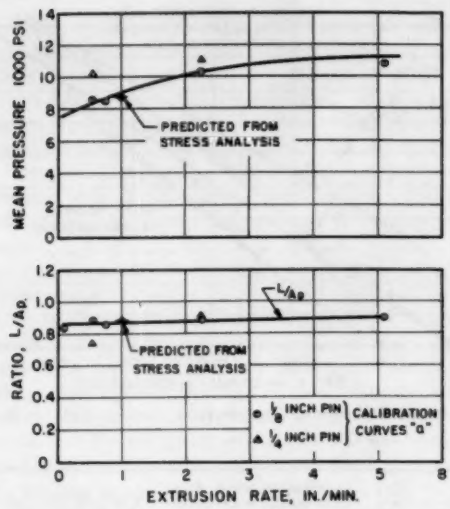

FIG. 8 MEASURED AND PREDICTED MEAN PRESSURE AS A FUNCTION OF EXTRUSION RATE
(Distance between pressure gages and die is 0.4 in.)

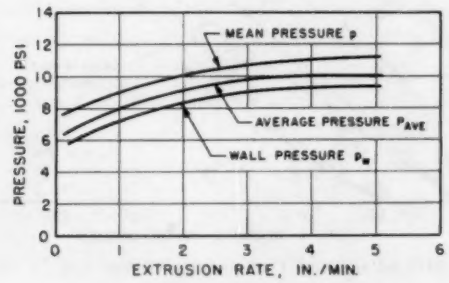

FIG. 9 COMPARISON BETWEEN MEAN PRESSURE, WALL PRESSURE, AND AVERAGE PRESSURE AS FUNCTIONS OF EXTRUSION TIME
(Distance between pressure gages and die is 0.4 in.)

CONCLUSIONS

1 The mean pressure within an extrusion billet can be determined experimentally with a pressure gage.

2 The size of the gage pins inserted in the wall of the extrusion cylinder has a definite influence on the accuracy with which the mean pressures can be found, the smaller pins giving more accurate values.

3 The same wall pressure was measured with different-sized gage pins and was found to agree with the calculated values.

4 The mean pressures found experimentally agreed with those predicted from a stress analysis based on the experimental flow patterns.

5 The applied load and wall pressure as well as the mean pressure were found to increase linearly with increasing extrusion rates.

ACKNOWLEDGMENTS

The author wishes to thank Prof. E. G. Thomsen of the Division of Mechanical Engineering at the University of California at Berkeley, for his valuable aid and numerous constructive suggestions. He also wishes to thank the College of Engineering for the use of its facilities and providing research grants for the investigation.

Discussion

E. V. CRANE.[8] This paper is a thorough presentation of the application and calibration of a strain-gage pressure pickup, with experimental data from the room-temperature extrusion of lead, to illustrate use of the device.

The data presented in chart form are interesting for purposes of interpretation. The shape of the curves obtained, illustrating change of pressure with increasing speed of extrusion, checks with experience in other thermoplastic metal working operations.

Since our knowledge of physical behavior of metals in their thermoplastic range is rather inadequate, the author's data and method together with cross-referenced work of Prof. E. G. Thomsen promise future advances. May we review the situation briefly in order to draw a further interpretation and raise a question for possible continued research?

Rubber, in an elastic noncrystalline state, acted like a hydraulic fluid in transmitting pressure from the direction of the applied force to the perpendicular direction of the measuring instrument in so far as could be detected in the data shown in Figs. 3 and 4 of the paper.

Lead, a metal, did not so act as shown in Figs. 6 and 7. Its interatomic bonds are still too potent to permit the fluid behavior which it would have at a higher temperature (above its melting point). Neither does it have the characteristics of the crystalline state which exists at lower temperatures where metal is elastic up to stresses described as elastic limit and yield point.

The valuable range is that in which most of the mass-production utility of plastically worked metals may be realized. In that range we have the phenomena of slip-plane movement, work-hardening, and increasing yield point. Poisson's ratio enters into directional relationships in some way which does not seem too clear as we pass from the elastic state to crystoplastic "cold" flow.

The recrystallization temperature range of the annealing process, varying with time and prior working, separates crystoplastic behavior from thermoplastic behavior. This is the technical boundary between the loose trade terms of "cold" working and "hot" working, forging, etc.

In the thermoplastic state both experimental work and observation shows that work hardening takes place even though an annealing readjustment, dependent upon the thermal energy of the atoms, is tending to return interatomic forces to "unstrained" balance. The directional effect or "polarity" of such forces may enter into our present problem. Time, temperature, and space lattice relationships are all part of it.

Lead is in its thermoplastic range at room temperatures, a point which the writer believes should be emphasized, for its implications should influence those who would make practical application of the valuable work in this paper.

Figs. 6 and 7 of the paper combine the results of thermoplastic behavior of lead with dimensional characteristics of the test apparatus.

Curve *a* rises with some of the characteristics of the stress-strain curve of the crystoplastic state but without a defined elastic limit or yield point. The lead billet deforms thermoplastically until it fits and fills the test chamber. Then the stress-strain relation expressed by curve *a* parallels the rubber calibration curve as the applied force expands the container elastically. It is higher, however, showing the need for more applied "axial" force to produce the "radial" force and movement detected by the SR-4 strain gage when calibrated with rubber.

Here we have characteristics of thermoplastic pressure-tem-

[8] Chief Engineer, E. W. Bliss Company, Canton, Ohio; Chairman, ASME Research Subcommittee on Plastic Working of Metals. Mem. ASME.

perature directional relationships. It is distinctly neither a hydraulic fluid relationship nor a crystalline elastic one. There is some space lattice angular transmission problem as yet ill defined in the art.

The return curve *b* in Figs. 6 and 7 shows that the test develops a prestressed state in which the container remains under elastic stress and the lead holds at a temporary "solid" yield point which might be interpreted at about 1600 psi, from the rubber calibration curve. Actually, it is higher owing to the loss of pressure in the right-angle transmission through lead. This higher apparent yield stress might be averaged at about 2800 psi by extending the curves *a* until they intersect the zero stress scale.

The repetitive curves *c-d* show in an interesting manner the combined effect of container elastic stress with the thermoplastic characteristics of the lead. As direction is reversed the container stress alternately opposes and aids applied load, resulting in the "hysteresis" loop effect. The residual strain reading of about 45 microinches on the gage would probably "creep" back to zero with sufficient time, dependent upon the thermoplastic temperature-creep characteristics of lead.

May we argue that curve *c*, extended back to zero in Figs. 6 and 7, represents a proper calibration between radial gage reading and the axial stress in the lead near the gage, and that this calibration may be applied to the actual working conditions during extrusion? If this is proper, as it seems to be, then the wall pressure may be replotted to the axial calibration by proportion to the rubber calibration line. This was done in Fig. 10, herewith, and the resultant dotted curve checks clearly with Professor Thomsen's theoretical derivation of the local pressure in that area.

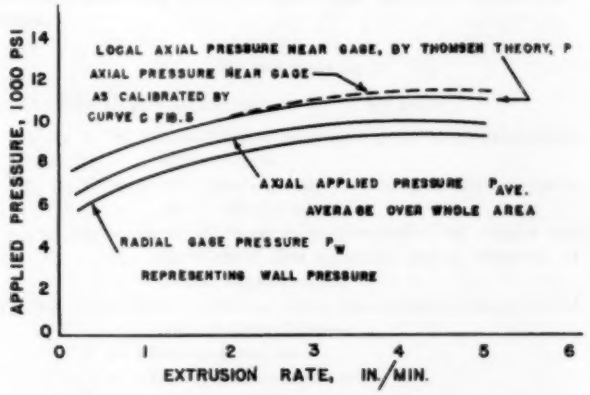

FIG. 10 AUTHOR'S FIG. 9 WITH DOTTED CURVE ADDED

(The local pressure on "trapped" lead in corner by gage is properly higher than average of pressure over whole area, including orifice. Does approximately 20 per cent difference between radial and axial stress in lead in that area represent a nonfluid directional ratio for thermoplastic lead, on the order of Poisson's ratio for crystalline state?)

The fact that the applied pressure (axial) is lower is logical because it is the average of pressure applied over the whole area compared with a high local pressure in a trapped corner. The difference merely emphasizes the high "viscosity" or interatomic "internal friction" of thermoplastic lead. (Local axial pressure in the orifice is obviously below the average.)

The radial stress in the semistatic corner metal near the gage appears to be about 20 per cent less than the axial stress, at the different extrusion speeds. May we theorize that this traces to some remaining directional difference in interatomic forces in the thermoplastic state? Does the 20 per cent figure represent something akin to Poisson's ratio for thermoplastic lead? Does it vary with temperature? These may be proper questions for further research into thermoplastic flow properties of metals.

AUTHOR'S CLOSURE

The author is indebted to Mr. Crane for his valuable comments. The question is raised whether curve *c* rather than curve *a* in Figs. 6 and 7 should be the proper calibration curve of the pressure gage. Computations on the stress analysis of tubular lead extrusions which have been done since the presentation of this paper showed an agreement within approximately 5 per cent between predicted and measured wall pressure when curve *a* in Fig. 6 is used.

The range of wall pressures for lead extrusion is between 8500 and 10,000 psi. A comparison of the microinch readings of curves *a* and *c* for any given stress level within the foregoing range shows a difference which is so small one may consider it to be almost within the realm of experimental error. The dotted curve of Fig. 10 shows the small change involved when either curve *a* or *c* is used in that range.

The discusser's comparison of curve *a* in Figs. 6 and 7 to the stress-strain curve is very interesting. If curves *b* and *c* are similar to the unloading and reloading curves of a stress-strain diagram after the plastic range has been reached, then the analogy could be carried further, namely, that curve *a* is representative of wall pressures during the initial extrusion run but curve *c* gives wall pressures for runs which take place after interruption and subsequent reloading.

The suggested study of the relationship between average pressure and radial wall pressure as a function of temperature should yield some very interesting and pertinent data; however, the presence of thermal stresses in the pressure gages would necessitate a complete change of the present design.

A New Approach to Metal-Forming Problems

Experimental Stress Analysis for a Tubular Extrusion

By E. G. THOMSEN,[1] BERKELEY, CALIF.

An incremental inverted extrusion process was used to obtain magnitude and direction of particle velocities in a billet during the pressing of a tubular product. The material chosen was commercially pure lead which can be hot-worked at room temperature and is believed to be a representative species of hot-worked metals. The reported graphical technique using experimental data is applied to a problem presently insolvable by analytical methods. It was found that both axial stresses and mean pressures in the region of the die were higher on the extrusion axis and near the cylinder wall than in the central region of the billet. This stress distribution is appreciably different from that of a solid extrusion for the same reduction in area. The present solution is a distinct advance in the direction of understanding three-dimensional metal-forming problems.

Nomenclature

The following nomenclature is used in the paper:

$\sigma_z, \sigma_\theta, \sigma_r$ = normal true stresses at a point in cylindrical co-ordinates, psi

$\dot{\epsilon}_z, \dot{\epsilon}_\theta, \dot{\epsilon}_r$ = normal plastic strain-rate components associated with normal stresses

$\tau_{zr}, \tau_{r\theta}, \tau_{\theta z}$ = shearing stresses perpendicular to co-ordinate direction of first subscript and in direction of second subscript, psi

$\dot{\gamma}_{zr}, \dot{\gamma}_{r\theta}, \dot{\gamma}_{\theta z}$ = plastic shearing-strain-rate components associated with shearing stresses

$\bar{\sigma}$ = effective stress, psi

$\dot{\bar{\epsilon}}$ = effective plastic-strain rate

$\dot{\lambda}$ = plasticity modulus

u, v = particle velocity components in axial z and radial r-directions, respectively

α_s = angle between principal strain rate and extrusion axis

p_w = wall pressure measured with SR-4 gages, psi

p = mean pressure, psi $= -\frac{1}{3}(\sigma_z + \sigma_\theta + \sigma_r)$

K_z = axial stress-reference datum, psi

L = applied load, lb

Introduction

Many engineering problems are not readily amenable to solution by a purely analytical approach. The equations describing a particular physical phenomenon either become too formidable to solve, or not enough independent relationships can be found to satisfy all the unknowns. Problems of this kind are classified as indeterminate and may be found in all branches of engineering, fluid mechanics, heat transfer, structure analysis and forming of metals, to name just a few. Hence in these cases the engineer, seeking answers to specific problems, is obliged to invoke known or new experimental techniques to fill in the missing information not obtainable by mathematical analysis alone.

"Visioplasticity" is a newly developed technique for obtaining a complete solution by experimental means of a certain class of mathematically insolvable metal-forming problems. The method consists of making the motion of a plastic metal mass visual and determining particle velocity and direction. Under steady-state conditions this information suffices to solve by graphical integration the stress and strain-rate distribution within this plastically moving metal. The present paper briefly describes the method and gives the stress and strain-rate distribution within a billet during an inverted extrusion process of a tube using commercially pure lead. The reader is referred to an article in three parts[2] in which the step-by-step analysis of visioplasticity has been described for the extrusion of a solid lead bar.

The method of analysis to be described is quite general and can be used for a number of metal-forming processes such as drawing and rolling and for work-hardening and non-work-hardening metals, i.e., cold and hot forming of metals.

Particle Motion During Metal Forming

In order to determine the flow pattern of a plastically moving metallic mass it is necessary to tag particles within the metal and to follow them during the deformation process. This was done for the case of a solid cylindrical billet, to be extruded into a tubular product, by sectioning the billet along a meridian plane and providing the billet on this plane with intersecting grid lines. The method of extruding a tubular product from a solid billet is not conventional, but permits comparison with previous results obtained with a solid extrusion. The grid-line intersections were considered to be particles whose motion must be determined. Grid lines after various stages of deformation during an inverted extrusion process at room temperature using commercially pure lead are shown in Fig. 1(*a* to *b*) inclusive.

The direction and magnitude of velocity of particles were determined by subjecting the billet to a series of small deformation steps and noting the displacement of the particles during these steps. Since the meridian plane is hidden from the observer's view when in the extrusion apparatus, Fig. 2, it was necessary to disassemble the billet from the apparatus between each step. A permanent record of particle position between each deformation step was taken on a negative in a fixed photographic

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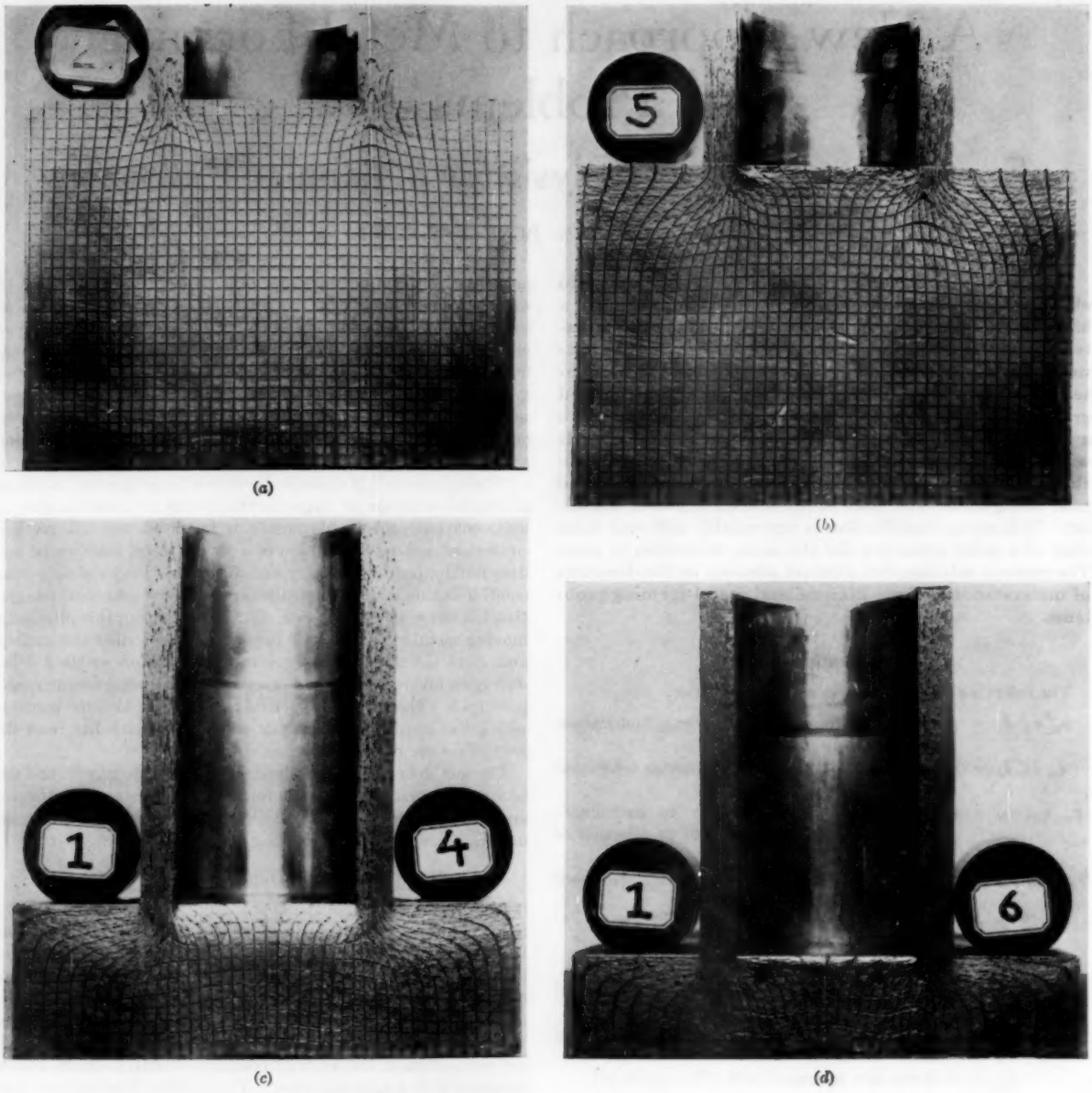
[1] Associate Professor of Mechanical Engineering, College of Engineering, University of California.

Contributed by the Metals Processing Research Committee and presented at a joint session of the Mexico Section and the Metal Processing Research Committee at the International Meeting, Mexico City, Mexico, March 10–12, 1954, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, December 30, 1953. Paper No. 54—Mex-16.

[2] "An Experimental Investigation of the Mechanics of Plastic Deformation of Metals," by E. G. Thomsen, C. T. Yang, and J. B. Bierbower, University of California Publications in Engineering, vol. 5, no. 4, pp. 89, et seq., University of California Press, Berkeley and Los Angeles, Calif., 1954 (in process of publication).



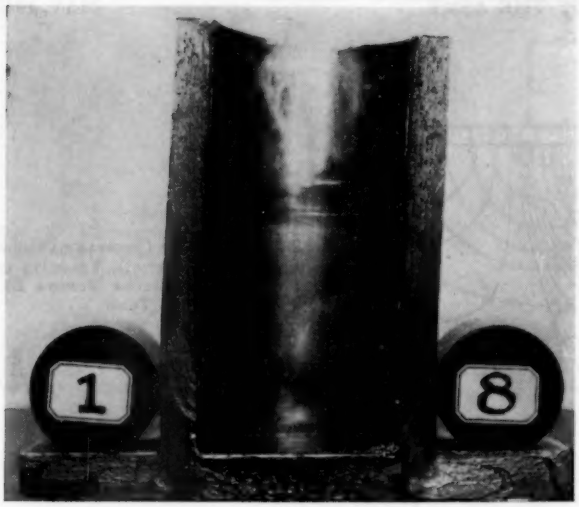
(a) (b) (c) (d)

FIG. 1(*a* TO *d*) LEAD BILLET IN SEVERAL STAGES OF DEFORMATION

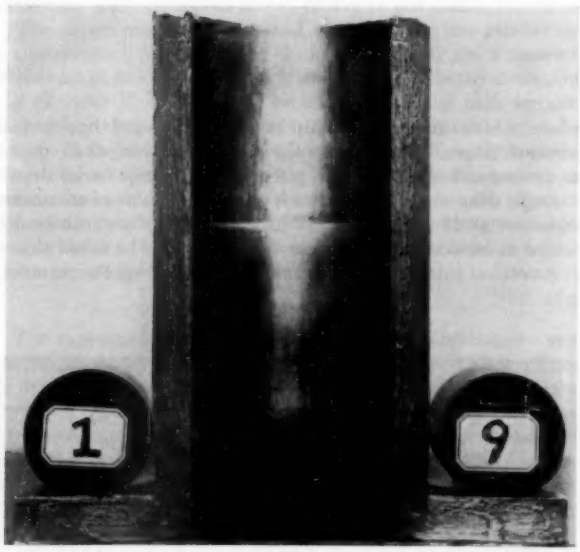
setup. These negatives were then enlarged on a screen to 10 times the actual billet size and the instantaneous intersecting grid lines were plotted on the screen.

The flow directions as determined by this method are shown in Fig. 3 which is a composite photograph of this enlarged plot. The actual extrusion steps from which the plot was constructed for this 4.3-in-diam billet extruded into a tube with outside and inside diameters of 2.15 in. and 1.52 in., respectively (reduction in area = 87.5 per cent), varied from approximately 0.025 in. to 0.10 in. change in billet length per step. The short steps of 0.025 in. were necessary to fix the direction of flow near the die opening, where the relative particle velocity is high as compared with that in the corner formed between the die plate and the wall of the extrusion chamber or the extrusion axis. This flow pattern is characteristic for this particular die shape and reduction in area. For a different die shape or a different reduction in area the flow pattern changes. This is strikingly portrayed in the flow pattern shown in Fig. 4 which was obtained with a 1.5-in-diam solid extrusion also pressed from a 4.3-in-diam lead billet at a reduction in area of 88 per cent.

In order to make a complete stress and strain-rate analysis it is necessary that in addition to their direction the magnitude of the instantaneous particle velocities be known. Hence the magnitudes of the particle velocities for the tubular extrusion are shown in Fig. 5 as trajectories of constant-velocity ratios. These ratios are dimensionless and were obtained by dividing the particle velocity as represented by average particle displacement per unit time during a small step by the velocity of the butt end of the



(e)



(f)

FIG. 1(e, f) LEAD BILLET IN ADDITIONAL STAGES OF DEFORMATION

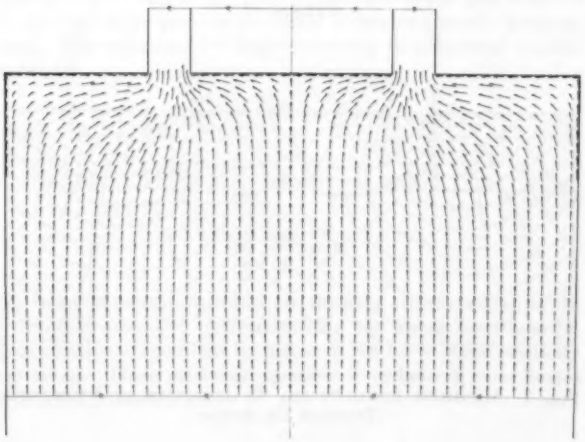


FIG. 3 (*left*) STEADY-STATE PARTICLE-FLOW DIRECTIONS, WHEN EXTRUDING A 4.3-IN-DIAM BILLET INTO A TUBE, WITH OD = 2.15 IN. AND ID = 1.52 IN.

(87.5 per cent reduction in area. Material is commercially pure lead extruded at room temperature using MolyKote lubrication.)

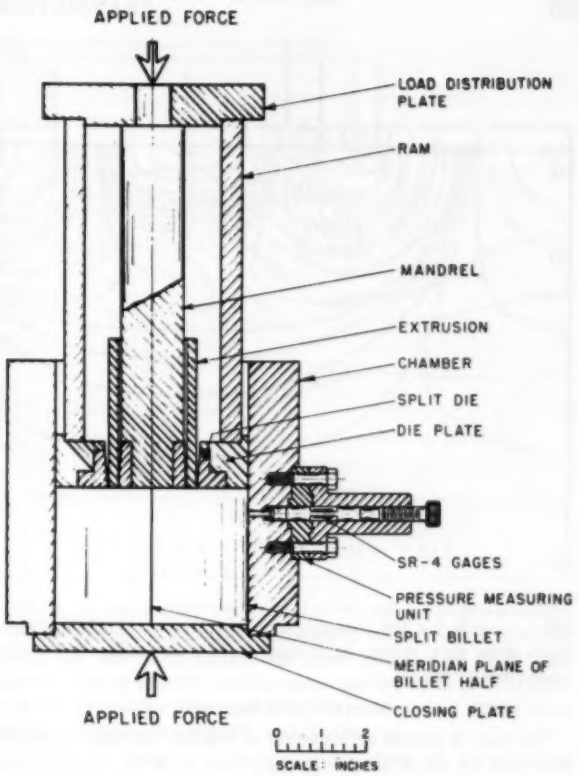


FIG. 2 SCHEMATIC DIAGRAM OF EXTRUSION APPARATUS

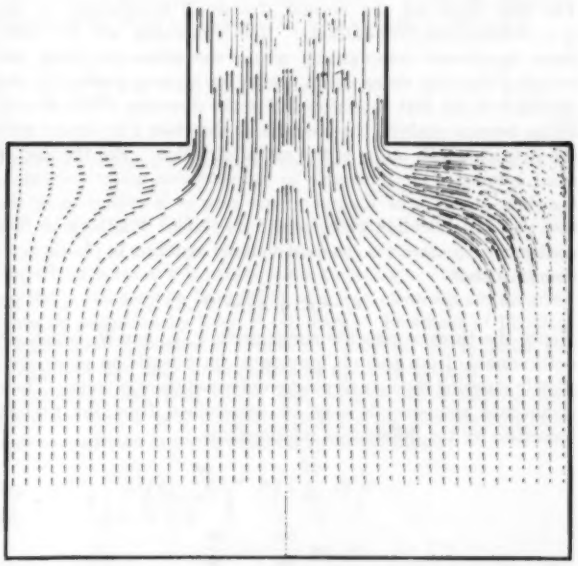


FIG. 4 STEADY-STATE PARTICLE-FLOW DIRECTIONS, WHEN EXTRUDING A 4.3-IN-DIAM BILLET INTO A 1.5-IN-DIAM SOLID BAR

(88 per cent reduction in area. Material is commercially pure lead extruded at room temperature.)

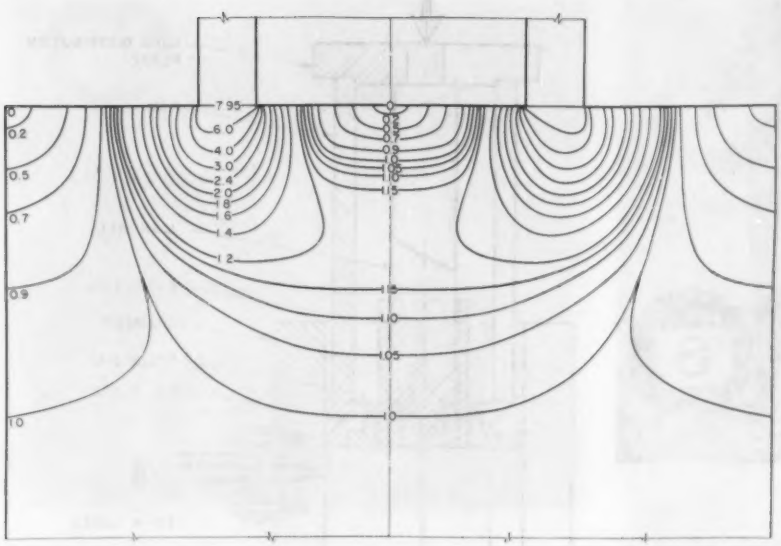


FIG. 5 TRAJECTORIES OF CONSTANT DIMENSIONLESS RATIOS OF PARTICLE VELOCITY TO VELOCITY OF END OF BILLET DURING EXTRUSION OF TUBE

billet, considering the movable die to be stationary. The trajectories were then drawn as average curves through many calculated velocity ratios.

STRAIN-RATE ANALYSIS

The rate of plastic deformation at a point may be completely described by six strain-rate components as given in cylindrical co-ordinates by

$$\dot{\epsilon}_z,\ \dot{\epsilon}_r,\ \dot{\epsilon}_\theta,\ \dot{\gamma}_{zr},\ \dot{\gamma}_{r\theta},\ \text{and}\ \dot{\gamma}_{\theta z} \qquad [1]$$

The first three are the normal strain-rate components in the z, r, θ-directions, respectively. The remaining are the three shear strain-rate components, where the subscripts have the meaning that the shear strain occurs on a plane normal to the direction of the first subscript and in the direction of the second. In the present analysis the co-ordinate direction z is parallel with the extrusion axis and r is the radial direction. For the case of axial symmetry as applicable to the extrusion under discussion, the shear-strain rates $\dot{\gamma}_{r\theta}$ and $\dot{\gamma}_{\theta z}$ are zero as is evidenced by the fact that the meridian plane remains a plane. Furthermore, the continuity equation (or constant-volume equation in plastic flow) demands that

$$\dot{\epsilon}_z + \dot{\epsilon}_r + \dot{\epsilon}_\theta = 0 \qquad [2]$$

Hence the three strain-rate components $\dot{\epsilon}_z$, $\dot{\epsilon}_r$, and $\dot{\gamma}_{zr}$ which are determinable from particle movement on a meridian plane completely describe the rate of plastic deformation at a point. These strain-rate components may be determined from the particle velocities by the differential relationships

$$\left.\begin{aligned} \dot{\epsilon}_z &= \frac{\partial u}{\partial z}, \quad \dot{\epsilon}_r = \frac{\partial v}{\partial r} \\ \dot{\gamma}_{zr} &= \frac{\partial u}{\partial r} + \frac{\partial v}{\partial z} \end{aligned}\right\} \qquad [3]$$

where u and v are the particle-velocity components in the z and r co-ordinate directions, respectively. These velocity components are readily determinable at any point on the meridian plane from the experimental data given in Figs. 3 and 5.

It is now of interest to establish the principal direction of the strain rates which correspond also to the principal stress directions where shear stresses are zero, if the plastic metal is assumed to be isotropic. This can be accomplished by substituting the experimental strain rates into the equation

$$\tan 2\alpha_z = \frac{\dot{\gamma}_{zr}}{\dot{\epsilon}_z - \dot{\epsilon}_r} \qquad [4]$$

where α_z is the angle between the extrusion axis z and the principal stress direction. Fig. 6 shows the calculated principal directions as crosses and also the faired principal stress trajectories drawn through these crosses. Inasmuch as the directions of maximum shear are at 45 deg to the principal directions, these can be obtained at once and are shown in Fig. 7. It may be noted that a symmetrical pattern is obtained and the metal along the extrusion

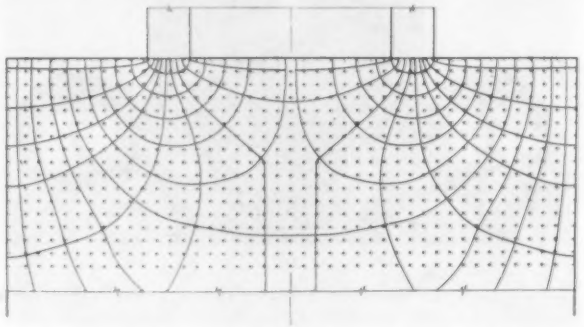


FIG. 6 PRINCIPAL STRESS TRAJECTORIES IN BILLET FOR TUBULAR EXTRUSION

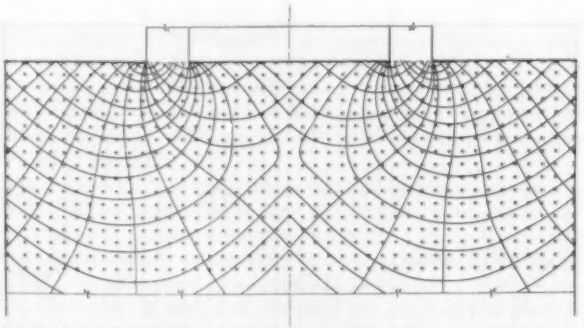


FIG. 7 MAXIMUM SHEAR-STRESS TRAJECTORIES IN BILLET FOR TUBULAR EXTRUSION

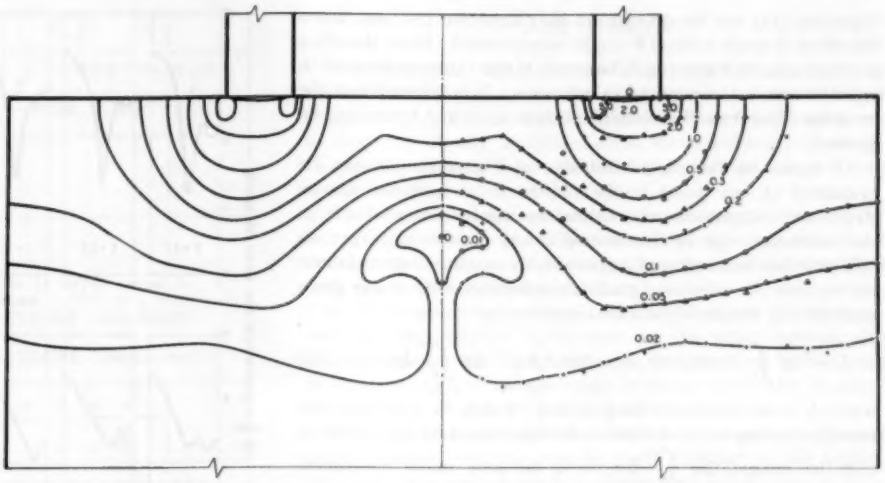


FIG. 8 CONSTANT EFFECTIVE STRAIN-RATE TRAJECTORIES IN BILLET FOR TUBULAR EXTRUSION

axis contains no shearing stresses as could be predicted, since the maximum shear directions are at 45 deg to the axis.

The experimentally determined constant effective strain-rate trajectories are plotted in Fig. 8 and numerically are a measure of the rate at which plastic work is being done. The significance of the effective strain rate will be discussed in the next section. It may be noted that the rate at which plastic work is being done is large in the vicinity of the die corners but significantly lower in the metal within the tube wall. This is of importance from a practical standpoint because it indicates that the rate of grain deformation along the surfaces of the extruded tube is appreciably greater than in the center. Hence nonuniform grain-size distribution may be expected in the extruded product.

STRESS ANALYSIS

The experimental analysis of the strain-rate distribution was carried out for steady-state conditions only. This applies as long as the billet length is at least 2.0 in. Steady-state conditions are disturbed, however, when the billet length becomes shorter than 2.0 in. because the stub end of the billet then enters the plastic zone, which extends approximately by this length into the billet upstream from the orifice. The effect of nonsteady-state flow may be seen by the initiation of the defect known as a pipe just starting in Fig. 1(*c*). This pipe, in subsequent stages of deformation, disappears and the central portion of the billet is sheared out as shown in Fig. 1 (*e*, *f*).

For steady-state conditions when the plastic metal is moving relatively slowly, in the range of commercially used extrusion speeds, the equations of static equilibrium apply and must be satisfied at every point in the metal undergoing plastic deformation. The equations for axial symmetry in cylindrical co-ordinates are

$$\left.\begin{aligned} \frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{zr}}{\partial r} + \frac{\tau_{zr}}{r} &= 0 \\ \frac{\partial \sigma_r}{\partial r} + \frac{\partial \tau_{zr}}{\partial z} + \frac{\sigma_r - \sigma_\theta}{r} &= 0 \\ \frac{1}{r}\frac{\partial \sigma_\theta}{\partial \theta} &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots [5]$$

These equations can be solved for the axial-stress gradient in the r-direction, $(\partial \sigma_z)/\partial r$, by transforming the stress Equations [5] into strain-rate equations by means of the Lévy-Mises plasticity equations. These plasticity equations in cylindrical co-ordinates with axial symmetry are

$$\left.\begin{aligned} \dot{\epsilon}_z &= \dot{\lambda}(\sigma_z + p) \quad \dot{\epsilon}_\theta = \dot{\lambda}(\sigma_\theta + p) \\ \dot{\epsilon}_r &= \dot{\lambda}(\sigma_r + p) \quad \dot{\gamma}_{zr} = 2\dot{\lambda}\tau_{zr} \end{aligned}\right\} \quad \ldots\ldots\ldots [6]$$

where σ_z, σ_r, and σ_θ are the normal stress components associated with the normal strain-rate components and τ_{zr} is the shearing stress, the subscripts having the same meaning as those used earlier for the strain-rate components; p is the mean pressure and $\dot{\lambda}$ is the proportionality factor between stress and strain rate, known as the plastic modulus, and is given by

$$\dot{\lambda} = \frac{3}{2}\frac{\dot{\bar{\epsilon}}}{\bar{\sigma}} \quad \ldots\ldots\ldots [7]$$

The effective strain-rate $\dot{\bar{\epsilon}}$ mentioned earlier, a complex quantity varying from point to point in the plastic mass, is given by

$$\dot{\bar{\epsilon}} = \frac{2}{3}\sqrt{\frac{1}{2}\left[(\dot{\epsilon}_z - \dot{\epsilon}_r)^2 + (\dot{\epsilon}_r - \dot{\epsilon}_\theta)^2 + (\dot{\epsilon}_\theta - \dot{\epsilon}_z)^2\right] + \frac{3}{4}\dot{\gamma}_{zr}^2} \quad \ldots\ldots [8]$$

The effective stress $\bar{\sigma}$ is a similar complex stress quantity given by

$$\bar{\sigma} = \sqrt{\frac{1}{2}\left[(\sigma_z - \sigma_r)^2 + (\sigma_r - \sigma_\theta)^2 + (\sigma_\theta - \sigma_z)^2\right] + 3\tau_{zr}^2} \quad \ldots [9]$$

The effective stress $\bar{\sigma}$ is proportional to Nadai's octahedral shearing stress and may be thought of as a root-mean-square shearing stress as suggested by Prof. E. R. Parker, Professor of Physical Metallurgy, University of California, since all terms under the square-root sign may be expressed as shearing stresses.

Assuming that lead obeys the Lévy-Mises equations and assuming that $\bar{\sigma}$ is a constant, then the equilibrium equations may be rewritten in terms of strain rates as follows

$$\frac{\partial \sigma_z}{\partial r} = \frac{2}{3}\bar{\sigma}\left[\frac{\partial}{\partial r}\left(\frac{\dot{\epsilon}_z - \dot{\epsilon}_r}{\dot{\bar{\epsilon}}}\right) - \left(\frac{\dot{\epsilon}_r - \dot{\epsilon}_\theta}{r\dot{\bar{\epsilon}}}\right) - \frac{1}{2}\frac{\partial}{\partial z}\left(\frac{\dot{\gamma}_{zr}}{\dot{\bar{\epsilon}}}\right)\right] \quad \ldots\ldots [10]$$

The effective stress for lead actually increases approximately linearly with strain rate as determined from compression tests[3] and follows the equation

$$\bar{\sigma} \cong \bar{\sigma}_0 + m\dot{\bar{\epsilon}} \quad \ldots\ldots\ldots [11]$$

In Equation [11], $\bar{\sigma}_0$ is an equivalent yield stress in compression or tension and m is the slope of the stress-strain-rate curve.

Equation [11] can be substituted into Equation [10] and hence the effect of strain rate on $\bar{\sigma}$ can be incorporated. Since the effect of strain rate on $\bar{\sigma}$ was small, however, it was taken as constant in order to reduce the required calculations. It is believed that the error resulting from this simplification is small and hence may be ignored.

All terms on the right-hand side of Equation [10] can be evaluated at any point in the plastic metal from the known strain-rate components and hence the axial-stress gradients in the r-direction can be determined at any position z. Equation [10] can then be integrated graphically by constructing continuous curves from the calculated gradients as functions of r at any given position z by means of the following equation

$$-L = 2\pi \int_0^{r_0} (-\sigma_z) r \, dr = -2\pi r_0^2 K_z - 2\pi \int_0^{r_0} \Delta\sigma_z r \, dr \quad [12]$$

where L = the total axial load, $\sigma_z = K_z + \Delta\sigma_z$, K_z = an unknown constant having units of stress to be determined for any section z, and the integral $2\pi \int_0^{r_0} \Delta\sigma_z r \, dr$ is the area under the curves constructed from the stress gradients of Equation [10]. It is assumed, of course, that the load is constant at any given section.

This is substantially true for the inverted extrusion process, since the friction loss appears to be small as demonstrated by the fact that the extrusion load remained nearly constant whether using lubricated or dry billets. If assumption of constant load is not permissible, the equilibrium equations can be rewritten and solved for the mean pressure gradients. Thus, if a pressure measurement can be made along the chamber wall by means of a pressure gage as used in the present experiments, see Fig. 2, then the equilibrium equations can be solved in a similar manner without requiring knowledge of the axial load distribution as a function of z.

Equation [12] can now be solved for the unknown K_z and hence the axial-stress distribution can be determined. The effective stress for lead is approximately 2000 psi and the axial extrusion load at an extrusion speed of $^1/_2$ ipm is 118,000 lb. With these values substituted the graphical integration yields the axial-stress distributions given in Fig. 9 for several z-positions from the die orifice.

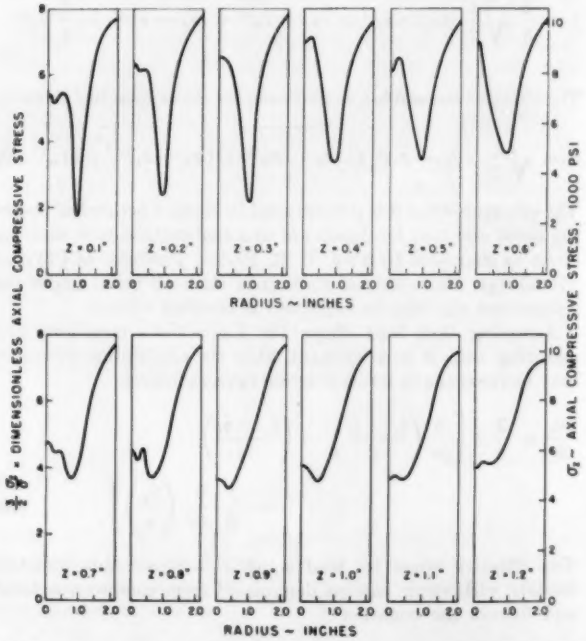


FIG. 9 AXIAL STRESS DISTRIBUTION IN BILLET DURING EXTRUSION OF TUBE OF COMMERCIALLY PURE LEAD AT ROOM TEMPERATURE AND A RATE OF $^1/_2$ IPM
(z = upstream distance from orifice.)

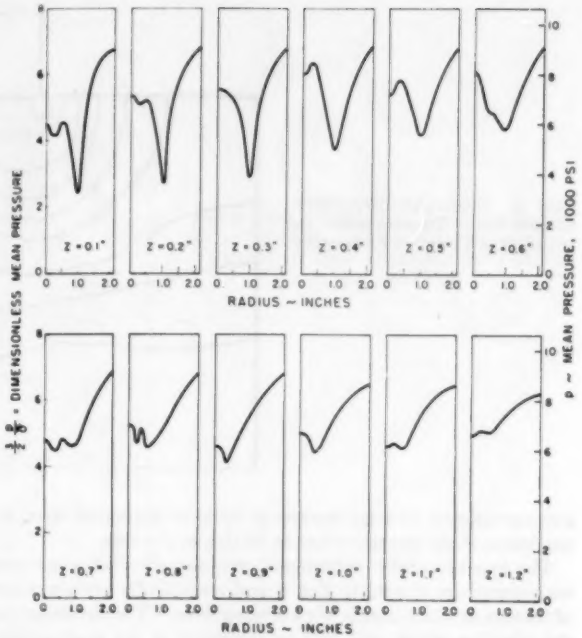


FIG. 10 MEAN PRESSURE DISTRIBUTION IN BILLET DURING EXTRUSION OF TUBE OF COMMERCIALLY PURE LEAD AT ROOM TEMPERATURE AND A RATE OF $^1/_2$ IPM
(z = upstream distance from orifice.)

It is of interest to note that the axial stress is highest at the cylinder wall and on the extrusion axis near the orifice. The steep stress gradients within the billet, of course, are the reason that the forming load, calculated by simple theories assuming uniform deformation, are in error by a factor of 2.

The mean pressure p also can be calculated from the plasticity equations as soon as σ_z is known, and the distributions are shown in Fig. 10. Similar steep pressure gradients also may be observed here but with the numerical values differing at the center and at the wall from those of the axial stresses.

CHECKING MEAN PRESSURE

It is now of interest to check the calculated mean pressures at the wall with those measured by the pressure gage attached to the extrusion chamber, previously referred to and discussed by Frisch and Thomsen,[3] and Frisch.[4]

The results are given in Fig. 11. The open circles are the wall pressures as determined from the new straight-line calibration curve obtained by Frisch and shown in Fig. 3 of his paper.[4] The corresponding measured mean pressures (closed circles) were taken from curve "C" of Fig. 6 of that paper, which this author believes to be the appropriate calibration curve. The crosses are the predicted mean pressures at the wall as calculated from the stress analysis, see Fig. 10.

[3] "An Experimental Study of Metal Extrusions at Various Strain Rates," by J. Frisch and E. G. Thomsen, Trans. ASME, vol. 76, 1954, pp. 599–606.

[4] "A Contribution to the Knowledge of Pressure Measurements During Metal Deformation," by J. Frisch, published in this issue, pp. 509–514.

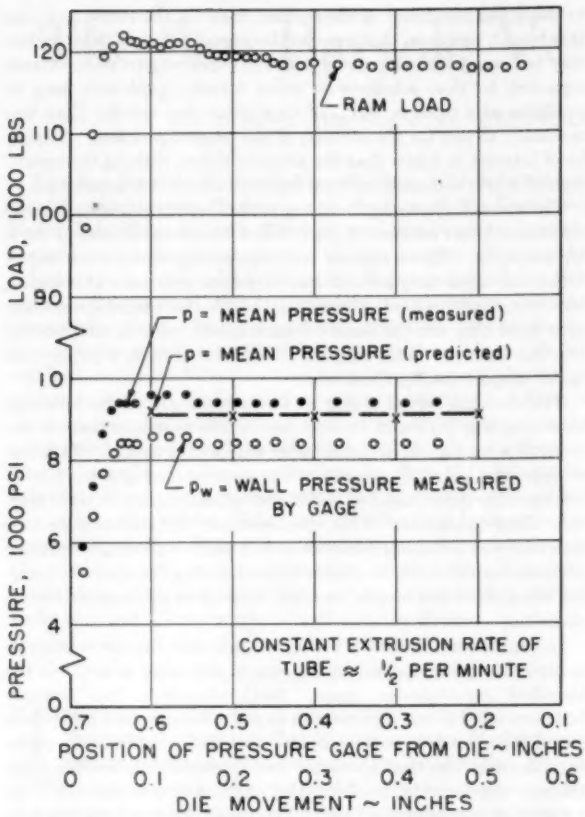

LOAD, 1000 LBS
PRESSURE, 1000 PSI
RAM LOAD
p = MEAN PRESSURE (measured)
p = MEAN PRESSURE (predicted)
p_w = WALL PRESSURE MEASURED BY GAGE
CONSTANT EXTRUSION RATE OF TUBE ≈ ½" PER MINUTE
POSITION OF PRESSURE GAGE FROM DIE ~ INCHES
DIE MOVEMENT ~ INCHES

FIG. 11 AXIAL LOAD AND PRESSURES AS FUNCTIONS OF DIE DISPLACEMENT DURING EXTRUSION OF TUBE OF COMMERCIALLY PURE LEAD AT ROOM TEMPERATURE AND A RATE OF 1/2 IPM

The close agreement between measured and calculated mean pressures is at once evident, and appears to justify confidence in the stress analysis developed herein.

CONCLUSIONS

1 The flow pattern on the meridian plane of a split cylindrical billet of commercially pure lead extruded at room temperature into a tubular product with a reduction in area of 88 per cent has been determined.

2 Particle-flow patterns in billets of commercially pure lead extruded into solid bars and tubes are substantially different for the same reduction in area.

3 The calculated axial compressive stresses and mean pressures in the billet during extrusion of a tube are highest on the extrusion axis and the chamber wall in the region near the die orifice.

4 Calculated mean pressures were found to be in good agreement with measured pressures.

ACKNOWLEDGMENTS

The author gratefully makes acknowledgment to Mr. N. J. Ku, former graduate student at the University of California, and to Prof. J. Frisch of the University of California, for their aid in obtaining the experimental flow patterns. He also wishes to thank the University of California for providing shop facilities and research grants for carrying out this investigation.

Discussion

W. S. WAGNER.[5] This paper contains a valuable contribution to the understanding of the metalworking process. Nevertheless, it appears that more material is mentioned than will be apparent to those not closely associated with stress-and-strain analysis. In general, those in the metalworking industries are not intimately familiar with these theories.

The following evaluation tends to present points which in the writer's opinion will add to the value of the paper and should not be construed to detract in any way from the material which has been presented by the author.

In the nomenclature, units have been given only for the stress and the applied loads. The units for the other symbols also should be given even though they may be "standard." The units in which values are expressed quite often aid materially in grasping the physical conceptions being described.

While symbols and equations used in the discussion of strain rate and stress analyses may be quite familiar to the author, they may be equally foreign to persons not using them as tools from day to day. A more explicit discussion of symbols and equations and their practical meanings would be extremely helpful in evaluating the results on a practicable basis.

An extension of the Bibliography to cover the origin of the equations used would help those who are unfamiliar with them to review their sources and derivations.

The report states that the pressure-gage readings do not agree with the mean hydrostatic pressure at the wall. The gage illustrated cannot be expected to read the mean of the pressures from the three co-ordinate directions. It will read only the wall pressure σ_r; will it not? Both mean pressures and axial stresses at the wall may be determined from Figs. 9 and 10. If these be known, the mean $(\sigma_\theta + \sigma_r)/2$ of the two stresses which lie in the plane normal to the axis may be calculated. Values of this mean are in the neighborhood of 8500 psi and more nearly approach the gage reading than does the mean hydrostatic pressure, $p = 1/3\,(\sigma_z + \sigma_\theta + \sigma_r)$. Is it not also conceivable that σ_r is less than σ_θ because σ_r is relieved by metal flow toward the axis or orifice? Can this stress, σ_r, in the radial direction be calculated from the data on which the report is based? It is the most important stress from the point of view of die design in order to determine bursting stresses in the chamber wall.

Figs. 1(*c*, *d*, *e*, and *f*) are included with little discussion except that they do not represent steady-state conditions. Is it possible to develop a general discussion of conditions which obtain in this state? The report states that the central portion of the billet was sheared out as shown in Fig. 1 (*e*, *f*). Was this "shearing" accomplished by advancing the mandrel while holding the position of the die plate?

Can values be assigned to the stress trajectories of Figs. 6 and 7?

The paper states that the method described may be used for both work-hardening and non-work-hardening metals. Lead is in its thermoplastic range at room temperatures; hence it falls within the non-work-hardening category. It would be interesting to see the results of such an experiment on an aluminum billet which would be in its crystoplastic range at room temperatures and therefore would be work-hardening.

AUTHOR'S CLOSURE

The author wishes to thank Mr. Wagner for his discussion of the paper and feels that he has raised many points which merit further clarification.

[5] E. W. Bliss Company, Canton, Ohio.

The material presented is admittedly in summary form in order to keep the paper within reasonable length, a publication practice of the Society. This has, naturally, the disadvantage that not all steps in the analysis are fully explained and hence can leave questions unanswered in the mind of a reader. That reader, for a more detailed discussion of the method of analysis, is referred to a series of articles cited under footnote 2, where a step-by-step description is given and to which a reasonably complete bibliography has been appended. This paper should appear in print during the latter part of this year, very probably before the present paper reaches the membership.

The strain rates are usually given without units since both shear and normal strain rates are involved. However, engineering units such as change in length per unit length per unit time (or in. per in. per minute) are implied when quoting the normal strain rates. The shear strain rates, on the other hand, refer to angular distortion and hence have dimensions of time (t^{-1}).

The mean pressure p is significant in three-dimensional plastic-flow problems since it appears in the plasticity equations (see Equations [6]). These equations show that the strain rates (or infinitesimal strains) are proportional to the difference in stress between a stress in the direction of straining and a mean tension stress which is given in terms of the negative value of the mean pressure. This difference in stress is known as a deviator stress and shows that plastic straining is not proportional to the normal stress but to this deviator stress.

This mean pressure p, as pointed out by Mr. Wagner, cannot be measured directly, but can be obtained by indirect means, as has been discussed in the papers cited under footnotes 3 and 4. If bursting pressures for the design of extrusion chambers are required, then it is necessary to know the magnitude of the radial stress σ_r. This stress can be obtained from the experimental data by calculation from one of the plasticity Equations [6], namely, $\dot{\epsilon}_r = \lambda(\sigma_r + p)$, since both $\dot{\epsilon}_r$ and p are known and the modulus λ can be determined. For the particular problem under discussion, the bursting pressure (σ_r) is given by the wall pressure p_w and was actually measured as shown in Fig. 11. It should be mentioned at this point that not all significant quantities of practical value have been shown, since the primary object of the experiment was to check the reliability of the method used for the stress analysis. It is hoped, however, that present theories of plastic flow of metals may be brought into harmony with the experimental observations reported, so that solutions to other forming problems may be available at a price in terms of time somewhat smaller than was necessary to pay for the solution of the present problem. It may be of interest to know that the analysis alone, without the experimental work, consumed several hundred hours of computation.

The stress trajectories for the principal normal stresses and the maximum shear stresses of Figs. 6 and 7 show directions of these stresses only. These stresses vary continually along these trajectories and hence they are not lines of constant stress. It is hoped, however, that these trajectories may be used for future stress analyses since they are the basis for such calculations in the Hencky slip-line theory.[6] This subject will be discussed in a paper now under preparation by the author.

The shearing out of a slug at the bottom end of the billet, as shown in Fig. 1 (*c* and *f*), was due to the unsteady state of deformation taking place in the latter stages of pressing. This was entirely due to readjustment of the stresses during this stage of deformation since mandrel and die were maintained in their relative fixed positions. This phenomenon of nonhomogeneous movement of the metal is also observed when extruding a solid bar where a central cavity or pipe is formed during the stages of forming when the billet length becomes very short as compared to its diameter.

In conclusion the author wishes to state that the stress analysis is presently being applied to the case of extruding a metal in the so-called crystoplastic range. 2S-O aluminum has recently been extruded in our laboratories at room temperature and it was found that the flow pattern is identical with that of lead, under conditions of identical geometry and degree of lubrication, even though the forming load for the aluminum was increased by a factor of approximately ten. The paper dealing with this subject was presented at the 1954 Annual Meeting of The American Society of Mechanical Engineers in New York, N. Y.

[6] "Über einige statisch bestimmte Fälle des Gleichgewichts in plastischen Körpern," by H. Hencky, *Zeitschrift für angewandte Mathematik und Mechanik*, vol. 3. 1923, pp. 241–251.

The San Bartolo Vertical Impulse Turbine

By JAVIER FUENTES,[1] SALVADOR ALMANZA,[2] AND W. J. RHEINGANS[3]

The San Bartolo Hydroelectric Station, which is part of the Miguel Aleman Hydroelectric System of the Comision Federal de Electricidad, Mexico, was designed to operate about six months a year, during the dry season. During the rainy season, water will be stored in the reservoir above this plant. Because of this intermittent operation, certain economies in the design and construction of the power plant were possible. The impulse turbine is a six-jet vertical unit which has certain advantages over a single-jet turbine, both in design and operating efficiency. Also, a vertical impulse turbine has a number of advantages over a horizontal unit, both in simplicity of design and in cost. The runner is of a unique design, in that it is integrally fabricated out of plate steel with the bucket bowls welded directly to the wheel disk, thus eliminating bucket bolts. A special needle-control mechanism permits operation of the turbine with any combination of six jets. The penstock shutoff valve consists of a rotary valve, which proved to be very satisfactory when tested in the shop.

The Reservoir System

THE San Bartolo No. 1 Hydroelectric Station is one of the power plants of the Miguel Aleman Hydroelectric System, which has been developed by the Comision Federal de Electricidad, using the water of the Balsas tributaries. Since this system has a rather novel water-distribution arrangement, a description of the principal features will be of interest.

The system is located in the Northwest of the Estado de Mexico and consists of the following plants: San Bartolo No. 1, San Bartolo No. 2, El Durazno, Ixtapantongo, Santa Barbara, and Tingambato (San Bartolo No. 2 is the only plant still under investigation; all the others are already either in operation or under construction).

The general arrangement of the Miguel Aleman Hydroelectric System is shown in Fig. 1.

The Villa Victoria Reservoir receives water from the Salitre River and feeds the San Bartolo No. 1 Plant, and in the future, the San Bartolo No. 2 Plant. San Bartolo No. 1 will have a total capacity of 28,980 kw in one unit, and San Bartolo No. 2 will have a maximum capacity of 20,700 kw. All the waters from several rivers, as Rio del Molino, Rio de Amanalco, and others, and the waters from the San Bartolo Plants, discharge into the Valle de Bravo Reservoir, which has a storage capacity of 400 million cu meters (324,000 acre-ft).

This reservoir feeds directly the El Durazno Power Station, which discharges into the Colorines Reservoir. The El Durazno Plant has a maximum capacity of 20,700 kw.

Paralleling the system described, we have the Tilostoc Reservoir which obtains water directly from the Malacatepec River and Valle de Bravo River.

This reservoir discharges downstream of the El Durazno Plant, directly to the Colorines Reservoir. This reservoir feeds the Ixtapantongo Hydroelectric Station, which will have a total capacity of 115,920 kw, discharging directly to the Ixtapantongo Reservoir which feeds the Santa Barbara Hydroelectric Station, with a capacity of 77,711 kw.

The water from Santa Barbara Plant discharges into the Santo Tomas Reservoir, which also receives the water from the Ixtapan de la Sal River.

The reservoir will feed the new Tingambata Station, which is under construction and which will have a maximum capacity of 155,250 kw.

Operating Conditions

The Villa Victoria Reservoir has a storage capacity of 210 million cu meters (170,000 acre-ft) having a surface of about 30.45 million sq meters (7500 acres) resulting in considerable evaporation; about 0.42 cu meter (14.8 cu ft) per sec in the summer. This is about 4½ per cent of the discharge of the San Bartolo unit under full load.

The Valle De Bravo Reservoir has less surface, which means that the evaporation is considerably lower. Therefore it is desirable during the summer to have as little water as possible in the Victoria Reservoir, in order to reduce the surface area and evaporation to a minimum. This is why most of the water should be transferred to the Valle de Bravo Reservoir which has a capacity of 400 million cu meters (324,000 acre-ft).

During the rainy season, the flow of the Colorines Reservoir from the Tuxpan and El Boxque Reservoirs and the discharge of the El Durazno Plant are enough to operate the Ixtapantongo and the other plants further downstream independently of the waters from the Villa Victoria Reservoir. Because of the foregoing factors, San Bartolo No. 1 (and San Bartolo No. 2) will have intermittent operation. Studies have shown that in order to take the best advantage of the storage and water of the Villa Victoria Reservoirs, it should be operated at a maximum capacity and its maximum efficiency during 6 months of the year.

General Description of San Bartolo No. 1 Station

This station receives 9 cu meters (318 cu ft) of water per sec from the Villa Victoria Reservoir, through 36,785 meters (22.7 miles) of canals, tunnels, and penstocks. The penstock is 2440 meters (8000 ft) long with a surge tank at the upstream end where it connects to a tunnel. The total net head for the plant is 376 meters (1233 ft).

Taking into consideration that this plant will operate only 6 months a year, it was decided to install only one unit for the total capacity of the plant. This unit will have only one bank of transformers and one transmission line, whereby the cost and maintenance are reduced to a minimum, also simplifying the remote-control operation.

The turbine will be an Allis-Chalmers vertical six-jet impulse turbine rated 39,000 hp, 376 meters (1233 ft) head, 428.6-rpm direct-connected to a 28,000-kva, 13,200-volt, 0.9 power factor, 3-phase, 50-cycle Allis-Chalmers generator with direct-connected exciter and pilot exciter. The generator is of the suspended type with a Kingsbury thrust bearing located above the rotor. The generator will have differential, low-voltage, over-

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[3] Manager, Hydraulics Section, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. ASME.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received by ASME Headquarters, January 7, 1954. Paper No. 54—Mex-9.

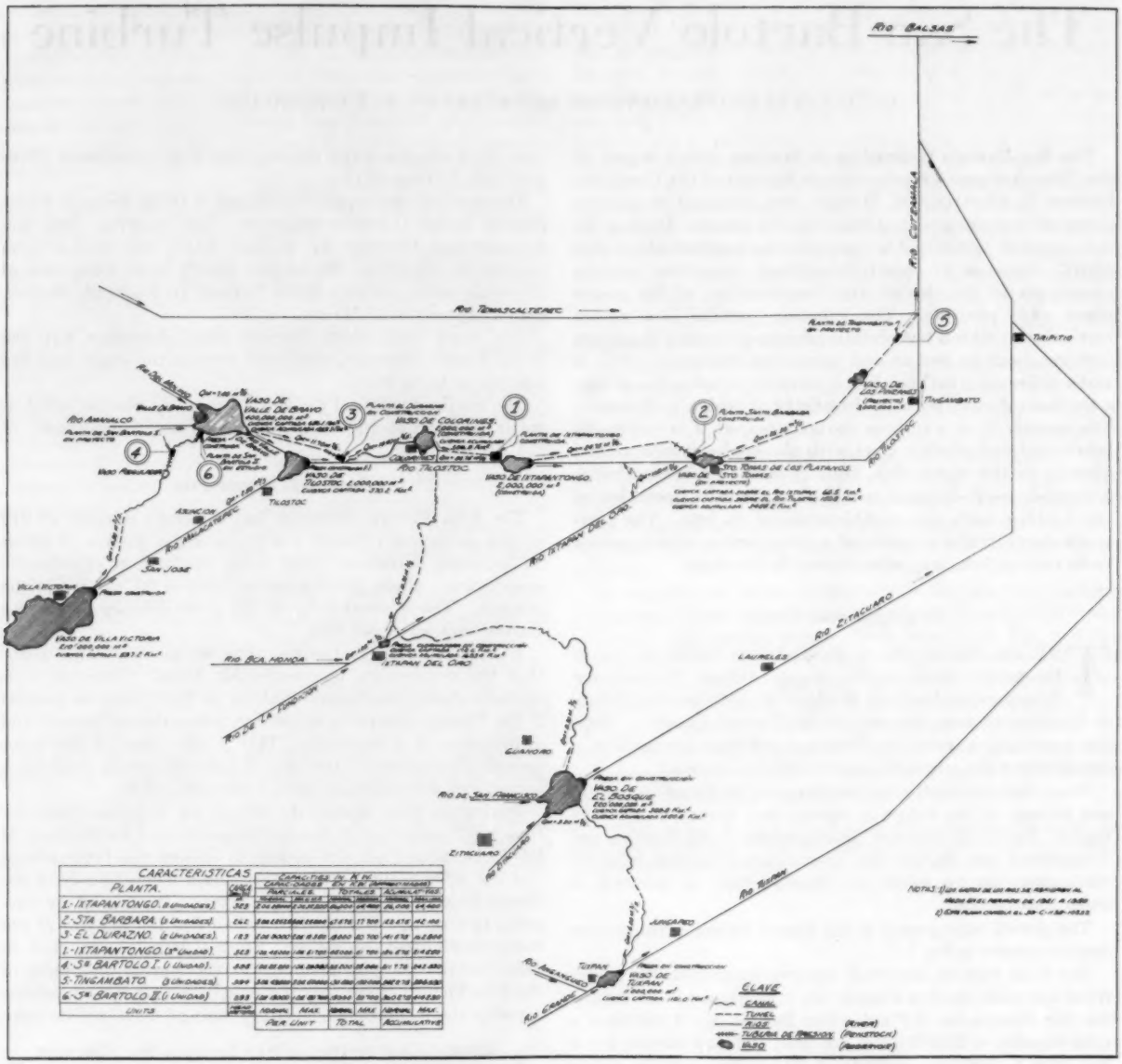

FIG. 1 GENERAL ARRANGEMENT OF THE MIGUEL ALEMAN HYDROELECTRIC SYSTEM

voltage, inverse-phase protection, and incomplete sequence of operation. There will be an Allis-Chalmers rotary-type shutoff valve in the penstock leading to the turbine. The unit will be regulated by a Woodward governor in combination with an Allis-Chalmers needle-control mechanism of special design.

This plant will have an outdoor substation, consisting of three single-phase transformers, OA type (self-cooled), of 9333 kva each, 13,200/86,600 volts, 50 cycles, connected in delta and star, with neutral ground.

From this substation, a line of 150 kv connects to the Miguel Aleman system which is interconnected with the Mexican Light and Power Company, which serves Mexico City.

REMOTE CONTROL OF PLANT

Owing to the special operating conditions it was decided that this plant should be operated locally and by remote control from the central station. Therefore all the necessary equipment to operate it by remote control was installed. There are three open lines between this plant and the control station; one will transmit the intermittent signals to start the operation of this plant; the second will transmit the telemeasure to the control board; and the third one will transmit continuously the level of the regulating reservoir above the San Bartolo Plant.

GENERAL TURBINE ARRANGEMENT

The San Bartolo turbine is a vertical-shaft, multiple-jet impulse wheel with six jets operating against a single runner. It has a single-turbine guide bearing located on the turbine head cover. The thrust bearing is located in the generator and was furnished as part of the generator equipment, Fig. 2.

The use of six jets on a single runner permitted the use of a relatively high specific speed (N_s — 11.6). Thus it was possible to use a comparatively small-diameter runner operating at a high speed. Naturally, the smaller the runner, the lower the cost of the

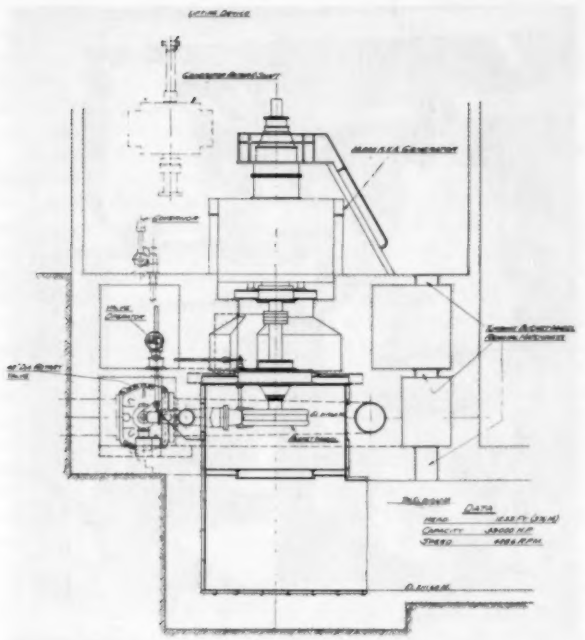

FIG. 2 ELEVATION OF SAN BARTOLO TURBINE

turbine. There was an additional saving in the powerhouse structure and excavation because of the small-size turbine.

The higher speed of the unit also resulted in a saving in generator cost. Therefore one of the advantages of a multiple-jet impulse turbine over a single-jet, or over a double-overhung horizontal unit, is the saving of the original cost of the turbine, generator, and powerhouse structure.

Another advantage of a multiple-jet unit is that if it is designed for operation with any combination of jets it can be operated at higher efficiency at part loads.

The unit is designed for full runaway speed which the rotating parts may attain with all six jets wide open under 376 meters (1233 ft) head and no load on the generator. Speed of the unit and pressure control in the penstock are maintained by a single governor and special needle-control mechanism.

GOVERNOR AND NEEDLE CONTROL

To avoid dangerous pressure rises and surges in the 2440 meters (3000 ft) of penstock between the surge tank and the turbine, the change of velocity of water under fluctuating load must be very gradual. To obtain quick response to load changes the output of the unit must be changed quickly by changing the water quantity impinging upon the buckets. One method of accomplishing this would be to control the impingement of the jets and to hold the flow-controlling needles stationary. This, however, is a water-wasting regulation and is prohibitive when water must be used economically.

It is therefore necessary that the six needles controlling the flow to the six jets be set to close and open in accordance with load demand, but at such gradual rates that no serious pressure rises or drops will be experienced. Accordingly, these needles are set to close at an average rate of 45 sec for the full flow and to open at an average rate of 40 sec, in order to limit the penstock pressure rise and drop. Such a slow closing rate would involve full runaway of the unit on sudden load rejection, and would result in inferior momentary load regulation. Therefore the quantity of water directed upon the buckets must be changed more rapidly.

In some installations this is accomplished by moving the needles rapidly and by opening a relief outlet as synchronously as possible to discharge the amount of water shut off by the needles, and thereafter closing the relief valve at a slow rate. However, a suitable relief valve with all the necessary controls, energy absorber, and so on, is an expensive piece of equipment.

A more practical and economical method, first introduced in 1928 in connection with the 60,000-hp impulse-wheel unit at Big Creek 2-A Plant of the Southern California Edison Company, is to provide means for diverting quickly the jet from the buckets and to let the flow-controlling needle follow the movement at a slower rate. This is accomplished by means of jet deflectors, actuated by the governor, which deflect the jets from the buckets in accordance with the reduction in load requirements. In the event of sudden load rejection, the jet deflectors can deflect the full-size jets in two seconds.

Since it was desirable to be able to operate the unit with any of the six jets alone or with any number or any combination of jets, a novel control system was furnished for the San Bartolo unit. Fig. 3 shows a schematic diagram of the governor and needle-control arrangement. The principal parts of this mechanism are as follows:

A standard gate-shaft governor with a flyball head driven electrically from a permanent-magnet generator.

Cut-in-type jet deflectors.

Needle and needle servomotor.

Special cam.

Special needle-control valves.

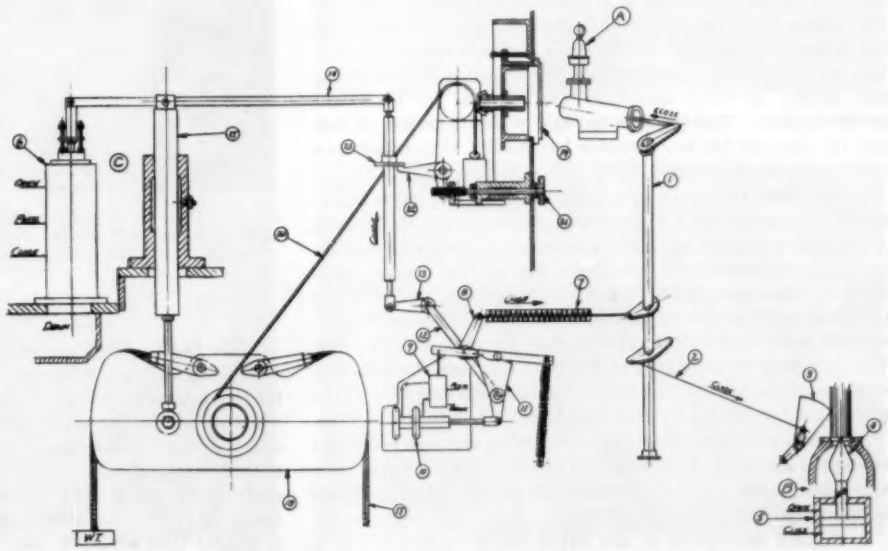

FIG. 3 SCHEMATIC DIAGRAM OF GOVERNOR AND NEEDLE-CONTROL MECHANISM

The general operation of this mechanism is as follows: The gate-shaft governor is directly connected to the six jet deflectors by a series of links and levers. The governor responds to a change in speed of the unit and moves the jet deflectors accordingly. If the speed of the unit increases, the jet deflectors will bite into the jet, deflecting some of the water from the buckets. The governor is of sufficient capacity to move the jet deflectors for their full stroke in 2 sec. Thus the full jet can be deflected from the buckets in 2 sec and thereby prevent excessive speed rise during sudden load rejection on the unit.

The needles which control the six jets are operated by individual servomotors. The movement of each servomotor piston, connected to the needle, is controlled hydraulically by an individual regulating valve, Fig. 4. These regulating valves are so arranged that they can limit the rate of movement of the servomotor pistons. Thus the movement of the needles, which control the flow in the penstock, can be set at a rate either closing or opening, which will prevent serious penstock pressure rise or pressure drop.

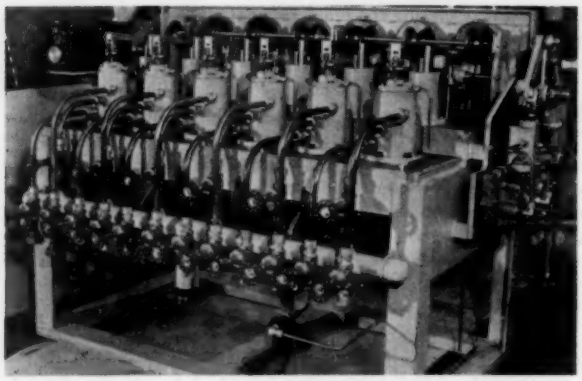

FIG. 4 REGULATING VALVES OF SPECIAL NEEDLE-CONTROL MECHANISM WHICH PERMITS USE OF ANY COMBINATION OF SIX JETS

The six needle regulating valves are operated by a single cam, Fig. 5, which in turn is operated mechanically by the gate-shaft-governor mechanism which also operates the six jet deflectors. The shape of this cam determines the relationship between the lips of the jet deflectors and the outer circumference of the jets. The use of the cam permits the jet deflectors to follow the jets closely, as their diameters are changed by movement of the needles. This permits the lips of the deflectors to bite into the jets with the least possible lost motion when called upon to do so by the governor.

A relay from the needle stem back to the floating levers on the needle regulating valves prevents overtravel of the needle. A needle-position indicator and a manually operated needle-control knob are provided for each needle, Fig. 6. The needle-control knob is so arranged that each needle can be set or limited to any predetermined opening or may be closed or kept closed entirely, without interfering with the operation of any of the other needles. This is a flexible arrangement for needle control whereby any combination of jets may be used. This permits operation of the unit at the highest efficiency at part load.

Although this mechanism was arranged so that each jet could be limited manually if so desired, it could be arranged readily to have each jet limited by a motor so that the individual jets could be controlled from a remote point.

A detailed description of this entire needle-control system is given in the Appendix.

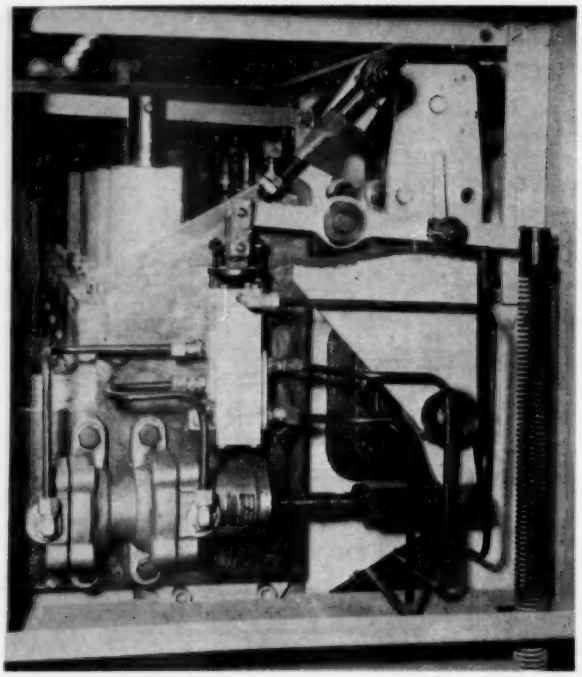

FIG. 5 SPECIAL NEEDLE-CONTROL MECHANISM SHOWING CAM OPERATED BY MOVEMENT OF GOVERNOR-CONTROLLED JET DEFLECTORS

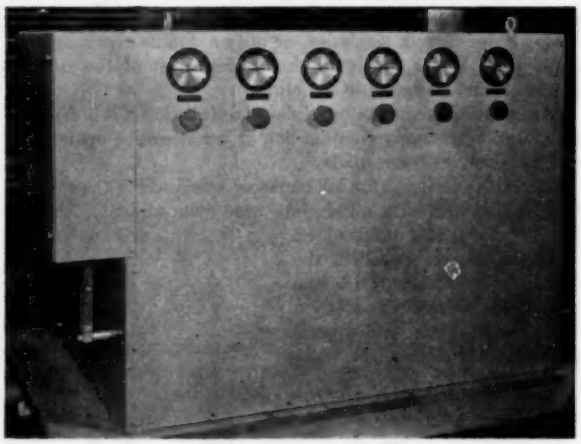

FIG. 6 CABINET FOR HOUSING SPECIAL NEEDLE-CONTROL MECHANISM WITH JET-POSITION INDICATOR DIALS AND KNOBS FOR MANUAL CONTROL OF INDIVIDUAL JETS

MECHANICAL FEATURES

Main Shaft. The main shaft is a steel forging, hollow-bored, with integrally forged flanges on each end. One end is arranged for coupling to the generator shaft. The other end is arranged for coupling to the runner. The shaft is 456 mm (18 in.) diam, which is larger than normally would be used for a Francis turbine of similar capacity and speed. The reason for this is that both the shaft and bearing are designed to take the unbalanced side thrust of one jet operating alone. On the other hand, the shaft is smaller than would be used for a horizontal impulse wheel of similar rating. For the horizontal unit, the shaft has to be de-

signed to take the overhanging weight of the runner, the unbalanced force of a single jet (with six times the force of a single jet on a six-jet unit), as well as the weight of the generator suspended between two horizontal bearings. Thus the vertical unit can make use of a relatively smaller-diameter shaft.

Bearing. The main guide bearing is of the babbitted, self-oil-lubricating type, mounted on the head cover. This type of bearing permits a simple oil-supply arrangement since it eliminates all oil pumps, motors, and oil piping and therefore is well suited for an automatic station. Here again, the vertical impulse unit has a decided advantage over the horizontal unit. The turbine guide bearing for a vertical unit can be made considerably smaller than the large heavy supporting bearings necessary for a horizontal unit. For example, a double-overhung impulse unit of this same rating would require two 560-mm (22-in.) diam × 1630-mm (64-in.) long bearings. In many cases such horizontal bearings are mounted on spherical seats for self-alignment of the shaft supporting the heavy weight of the runners and generator. Special oil-ring lubricating systems have to be provided, together with a high-pressure oil system for lifting the shaft so as to insure an oil film between the shaft and bearing while the unit is being started. All of this is unnecessary with a vertical unit.

Wheel Disk and Buckets. The wheel disk and buckets comprising the runner are of unusual construction. The bucket bowls were die-pressed out of plate steel in one piece with a forged-steel splitter welded in, Fig. 7. The bucket bowls were then welded to two solid plate-steel disks, or rings, with notches cut into them, so as to form lugs for supporting the bucket bowls, Fig. 8. The lugs to which the bucket bowls were welded thus formed an integral part of the wheel disk, eliminating bucket bolts and nuts.

Ribs were welded to the back of the bucket bowls to increase their stiffness. A solid one-piece plate-steel central hub was then welded to the two lug rings, Fig. 9. The hub was machined and drilled for attaching to the shaft flange.

After all the welding was completed the entire runner was furnace-stress-relieved. This type of construction could be called an integrally fabricated runner as distinguished from an integrally cast runner. However, it has the same advantages as an integrally cast runner, in that a larger number of buckets of a given size can be used for a given impulse diameter, without increasing the maximum lug stresses. This permits the use of higher specific speeds. Doing away with the bolting of individual buckets to the disk also eliminates the disk stresses which are set up by bucket bolts when installed with interference fits. In addition, the absence of bucket bolts and nuts reduces windage losses and reduces losses caused by interference with the discharge from the bucket bowls.

This is the first unit where this type of runner construction has been used. However, a similar integrally fabricated runner is being built by Canadian Allis-Chalmers, Ltd., for the 140,000-hp 762-meter (2500-ft) head four-jet vertical impulse turbine for Kemano Power Plant in British Columbia.

Nozzle Pipes. The intake pipe and nozzle pipe are annular, forming practically a complete circle with elbow take-offs for six jets, Fig. 10. It is in six sections with a take-off for each section. The

FIG. 7 DIE-PRESSED PLATE-STEEL BUCKET BOWL WITH WELDED-IN FORGED SPLITTER

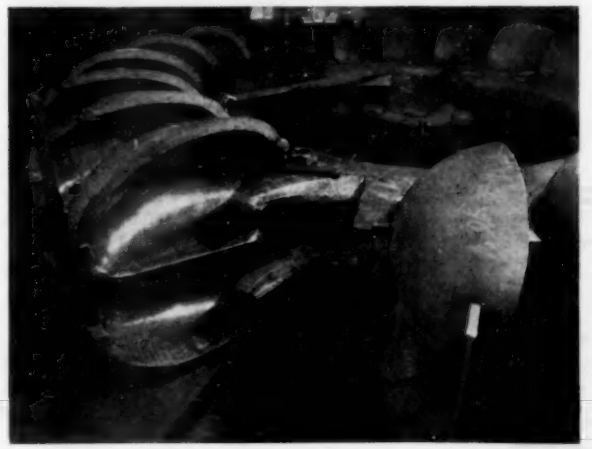

FIG. 8 ASSEMBLY OF BUCKET BOWLS TO TWO SOLID PLATE-STEEL LUG RINGS, PREPARATORY TO WELDING

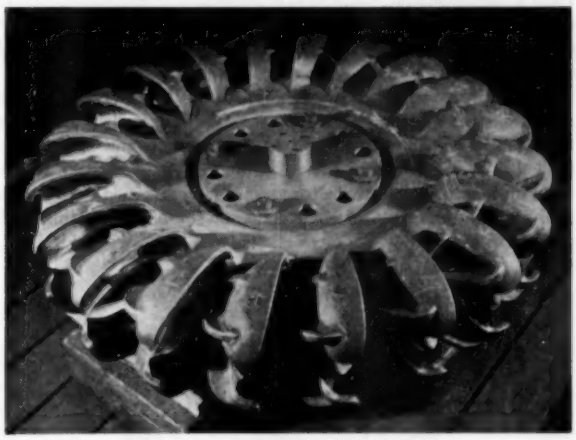

FIG. 9 COMPLETED INTEGRALLY FABRICATED PLATE-STEEL RUNNER

FIG. 10 INTAKE PIPE AND NOZZLE PIPE WITH SIX NOZZLES, SHOWING SEVERAL JET DEFLECTORS

take-off elbows are of cast steel welded to plate steel forming the nozzle pipe.

The inlet diameter of the nozzle pipe is 1.22 meters (48 in.) which corresponds to the diameter of the penstock rotary valve. In order to maintain uniform velocity throughout, the nozzle pipe is decreased gradually in size, whereas the take-offs for the six jets are all the same size. A nozzle is located at the end of each nozzle-pipe take-off, and is equipped with a renewable stainless-steel throat ring. Each jet is controlled by means of a needle with renewable stainless-steel tip. The needle stem is extended through the nozzle pipe to its outer periphery, where it is connected to an oil-operated servomotor, under control of a special control assembly connected to the governor system. The control assembly permits all needles to be operated either in unison or independently of each other. The entire needle-stem assembly, together with servomotor and piston, can be removed from the rear without disturbing the runner.

Jet Deflectors. The jet deflectors are of the cut-in type, which are mounted on the nozzle tip and are arranged to cut into the jet from one side. They deflect water from the buckets as soon as they cut into the jet. All the water which is cut off is deflected from the buckets. This is in contrast to the sleeve type, which has to move an appreciable distance into the jet before there is a noticeable reduction in load on the unit. The cut-in type, therefore, gives somewhat quicker and closer speed and load regulation. It also permits the nozzle tip to be set closer to the runner, thus reducing the distance the free jet has to travel before hitting the buckets. Since a free jet starts to spread and spray rapidly after leaving the nozzle orifice, reduction in the distance it has to travel before hitting the bucket increases the efficiency of the unit.

The entrance edge or knife-edge of the deflector is coated with stainless-steel weld, to reduce wear. All of the deflectors will be operated in unison by means of levers and links and will be controlled by the governor. They will be used for quick load reductions on the unit. The deflector linkage is so designed as to permit individual adjustment of each deflector for proper setting with respect to the jet.

Wheel Housing. The wheel housing is of ample size to provide room for the water to discharge from the buckets without interference or backsplash. It is made of welded-plate steel, suitably reinforced, and is arranged to support the head cover. A removable inspection platform is located about 6 ft below the center line of the runner. Provision has been made for removal of the runner by suspending it from the housing cover by means of eyebolts and lowering it to the bottom of the discharge pit. It can then be removed through hatchways provided for this purpose, Fig. 2.

Thus the runner can be removed without dismantling the generator or turbine. Energy-absorber plates are provided in the housing to absorb the full force of the jets in case of runaway, as well as when they are deflected from the buckets by the jet deflectors.

Wheel Pit. The bottom of the wheel pit is 6 meters (19.7 ft) below the turbine runner. At normal tail water there will be 3 meters (9.8 ft) of water in the wheel pit, which will act as an energy-absorber pool to absorb the discharge from the runner and the deflected jet when the jet deflectors are in operation.

Normal tail water is 3 meters (9.8 ft) below the turbine runner. This affords sufficient room to permit free discharge from the buckets and to prevent spray or backsplash from interfering with the runner. Although model tests at 61 meters (200 ft) head showed that there was no loss in efficiency of the unit with only 1 meter (3.3 ft) between the runner and the tail water, it is possible that the higher heads in the field will produce greater tail-water turbulence and backsplash. Therefore a greater distance was used.

The entire sides and bottoms of the wheel pit are lined with heavy steel plates to prevent damage to the concrete from the bucket discharge and from the deflected jets.

Head Cover. The head cover is of heavy plate-steel welded construction so designed that it forms a rigid support for the turbine guide bearing. It is arranged to be removable and is of such diameter as to permit removal of the bucket wheel upwardly through the generator. This affords an alternative method for runner removal.

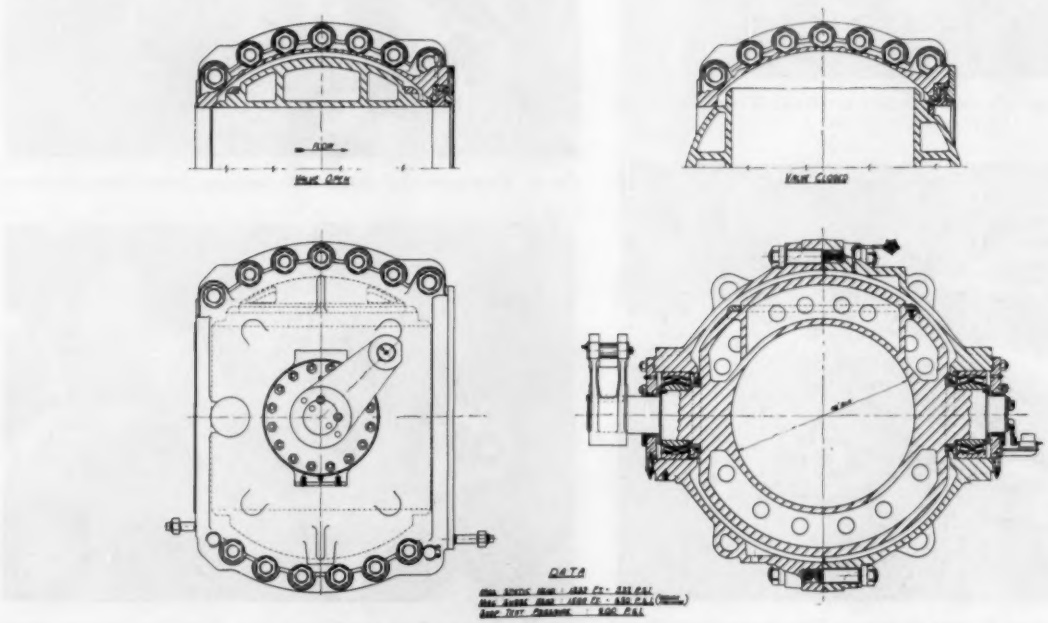

FIG. 11 DETAIL CROSS SECTION OF 1.22-METER (48-IN.) DIAM ROTARY VALVE

Materials. It is interesting to note that with the exception of the cast-steel nozzle-pipe elbows, the cast-steel jet deflectors, and the cast-iron bearing shell, the entire turbine is constructed out of plate steel. Modern methods of fabricating, forming, and welding have made this possible.

Rotary Valve. Fig. 11 shows a detail cross section of the rotary valve used as a penstock shutoff valve. The valve is 1.22 meters (48 in.) diam which is the diameter of the penstock and the inlet to the nozzle pipe at this point. Since there is no change in velocity through the valve, and no obstruction to the flow, the head loss through the valve is practically zero.

The valve is designed with a movable seal ring in the housing which seats on a spherical ring in the rotor, when in closed position. Movement of the seal ring is controlled by hydraulic pressure and the controls are so interlocked that it is impossible to move the seal ring to closed position until the rotor is in closed position. It is also impossible to move the rotor toward open position until after the seal ring has been unseated. Spherical roller bearings are used for the shaft trunnions to reduce rotating friction.

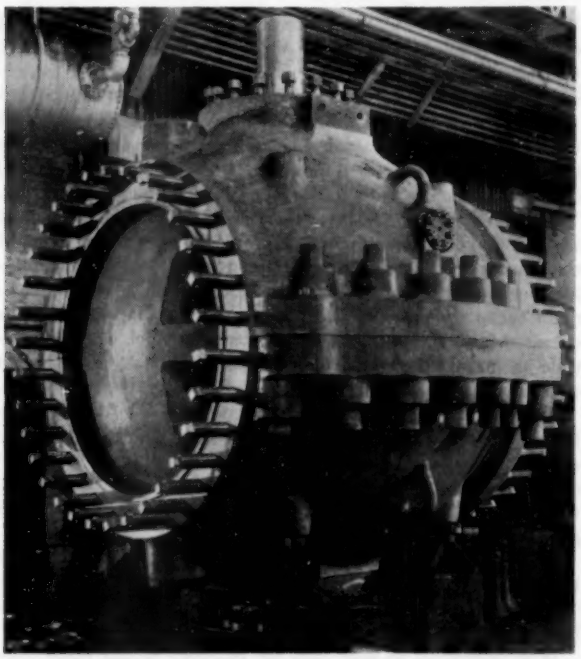

FIG. 12 SHOP ASSEMBLY OF 1.22-METER (48-IN.) DIAM ROTARY VALVE

The valve rotor is operated by a standard electric limit-torque operator, which uses an electric motor as a source of power. Fig. 12 shows the shop assembly of the valve. It was shop-tested hydrostatically under 900 psi. A leakage test on the valve seat was made under 550 psi. The total leakage from the seat and from the movable seal ring could be measured in drops per minute.

MODEL TESTS

Since this unit was not in operation at the time the paper was prepared, no field-operation performance data were available. However, model tests were made on an exactly homologous turbine, Figs. 13 and 14. The tests were made under 68 to 76 meters (225 to 250-ft head). They showed a maximum model efficiency of 89.2 per cent, when operating with 6 jets.

Fig. 15 shows guaranteed and model-test efficiency plotted against output. Model tests also were made with 4, 2, and 1 jets. The results are plotted in Fig. 15 and show how part-load efficiency can be increased by operating with fewer jets. For example, at 19 per cent load with only one jet operating, the increase in efficiency compared to six-jet operation is $4^1/_2$ per cent. This improvement is well worth while and justifies the expense of providing a control mechanism, Fig. 4, which makes this type of operation possible.

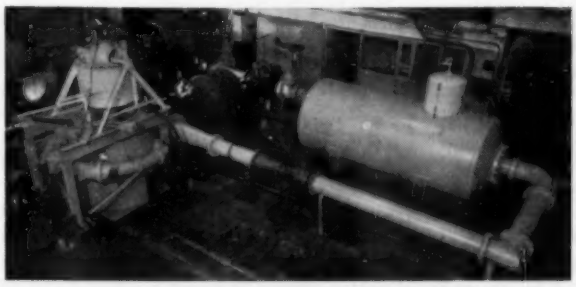

FIG. 13 TEST ASSEMBLY FOR MODEL OF SIX-JET VERTICAL IMPULSE TURBINE

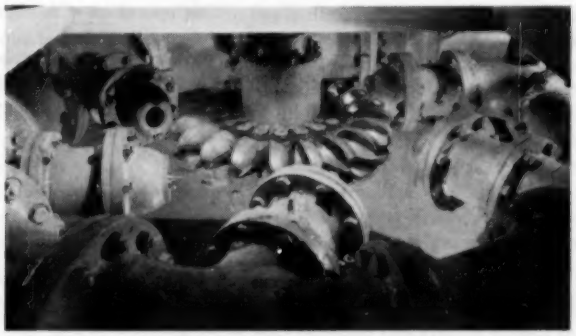

FIG. 14 MODEL NOZZLE-AND-BUCKET ASSEMBLY FOR SIX-JET VERTICAL IMPULSE TURBINE, MODEL TEST

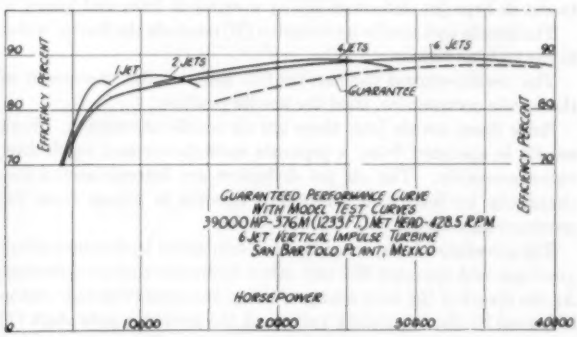

FIG. 15 GUARANTEE AND MODEL-TEST PERFORMANCE CURVES FOR SIX-JET VERTICAL IMPULSE TURBINE

(Model output stepped up to prototype size and field head.)

CONCLUSION

Because the San Bartolo Power Plant is fed by waters from a reservoir with a large surface area, resulting in high evaporation losses during summer months, the plant will be operated only during the winter months. During the rest of the year the reservoir will be drawn down to reduce evaporation losses. The

intermittent operation contemplated for this power plant made it possible to use economical design features in both the equipment and in the powerhouse structure.

Use of a six-jet impulse turbine made it feasible to go to a high unit speed, resulting in savings in both turbine and generator costs. Use of a vertical impulse wheel resulted in additional savings in costs. The high-speed unit also resulted in savings in powerhouse-foundation costs.

New design features such as an integrally plate-steel-fabricated runner and a special needle-control system were incorporated in the impulse turbine.

The integrally fabricated runner produced a sound structure with low operating stresses and minimum windage and friction losses. The special needle-control system which makes it possible to operate any combination of the six jets has distinct general operating advantages as well as permitting operation with high efficiencies at part loads.

The type of rotary valve used as a penstock shutoff valve has the advantages of practically zero friction losses and extremely low leakage losses. It is ideal for this type of installation where intermittent operation of the unit is being contemplated.

Appendix

Needle-Control Mechanism

Fig. 3 shows a schematic diagram of the governor and needle-control arrangement. The principal parts of this mechanism are as follows (letters and numbers refer to designations in Fig. 3):

The gate-shaft governor (A) has a flyball head which is driven electrically from a permanent-magnet generator mounted on the main generator. The governor gate shaft (1) is direct-connected to cut-in-type jet deflectors (3) by a series of links and levers.

The needle and needle servomotor (B) controls the flow of water to the turbine runner.

The needle-control mechanism (C) controls the movement of the needle servomotor, thus the needle position.

Since there are six jets, there are six needle assemblies. Each needle is operated from a separate servomotor and regulating-valve assembly. The six jet deflectors are interconnected mechanically by links and levers and operate in unison from the governor gate shaft.

The governor (A) is a standard self-contained hydraulic-turbine governor and operates like any other hydraulic-turbine governor. As the speed of the unit changes, the governor-servomotor piston is moved by the regulating valve and the governor gate shaft (1) is rotated. This moves the jet deflectors (3). When the speed of the unit is increased, because of a drop in load, the jet deflectors will bite into the jets, deflecting some of the water from the buckets. The governor is of sufficient capacity to move the jet deflectors for their full stroke in 2 sec.

Each of the needles (4) is controlled by a separate servomotor (5) which in turn is controlled by a separate regulating valve (6). A connecting rod (7) from the governor gate shaft is connected to the needle control. As the governor is operated in the opening direction, bell crank (8) is rotated counterclockwise. This will open valve (9) causing servomotor piston (10) to move to the right, thus rotating cam (11) counterclockwise. The roller follower and floating lever will recenter valve (9). The shape of cam (11) will determine the relationship between the outer circumference of the jet and the lip of the jet deflector (3).

As the cam (11) is rotated, shaft (12) will be rotated counterclockwise also. Lever (13) will rotate with shaft (12), causing the right-hand end of floating lever (14) to be lowered. The floating lever will rotate about its center which is pinned to the upper end of connecting rod (15). The regulating valve (6) will be opened and the needle piston of cylinder (5) will move in the opening direction. As the needle opens, a chain relay connection (17) will rotate relay segment (18) counterclockwise, and connecting rod (15) will be lowered. This will return regulating valve (6) to mid-position, and the opening motion of the needle will be stopped. Movement of the regulating valves is limited so as to limit the opening and closing rates of the needles.

A needle-position indicator (19) is operated by a chain (20) connected to the relay segment (18). Directly below each needle indicator is a small handwheel (21). When this is turned counterclockwise, stop (22) will engage collar (23). This will raise the right-hand end of floating lever (14) and move needle (4) in the closing direction. By turning the handwheel far enough, the needle can be closed entirely. Since there is a separate handwheel and needle-control valve for each jet, any needle can be set or limited to a predetermined opening or may be closed or kept closed entirely, without affecting the operation of any of the other needles.

Discussion

G. R. Woodman.[4] It is interesting to note that economy of operation requires draining the Villa Victoria Reservoir through the Bartolo Plants within a period of 6 months in order to save evaporation losses on this upper reservoir. This means that 49,600 kw of capacity at the Bartolo No. 1 and No. 2 plants that could be made available on a year-round stored-water basis is rendered idle for 6 months. Consequently, the over-all economy in the use of installed capacity is not readily apparent.

Turbine Equipment. In connection with the installation of the plant hydraulic equipment it would be interesting to know the reasons for selecting the impulse-type turbine in place of a lower-setting reaction type with its gain in both head and maximum turbine efficiency. It is assumed that the unit must be operated for long periods at partial loads in which case the multiple-jet impulse turbine could be more efficient by selective jet operation.

Considerable economy appears to be effected by the installation of the multijet vertical-type impulse turbine over the horizontal construction. Higher unit speed definitely reduces generating-equipment first costs and power-plant-structure costs, while the multijet operation produces high turbine efficiency over a wider load range.

The use of jet deflectors to eliminate relief valves and permit lower strength penstocks is obviously a saving in penstock costs. The Big Creek 2-A Unit mentioned has proved very successful both from an operating and maintenance point of view.

It is interesting to note that the penstock design takes complete advantage of the governor-deflector operating characteristics but the generator design adds a safety precaution against full runaway speed. We are in agreement with the precaution against runaway as we have experienced it at least twice during the past 20 years.

The erection of the unit to permit removal of the turbine runner from either the lower end or up through the generator will facilitate maintenance repairs and minimize repair costs.

Impulse Wheel. Integral fabricated water wheels are a fairly recent development in water-wheel construction that would appear to compete very favorably with cast integral wheels. It is important to secure sound welds in the fabricated integral construction. Consequently, design consideration facilitating and simplifying welding operations, together with welding skill and careful shop inspection, are extremely important to the success of this type of construction.

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[4] Southern California Edison Company, Los Angeles, Calif.

Rotary Valve. The use of the rotary valve for the lower penstock shutoff valve offers many advantages.

Minimum friction loss raises the over-all operating efficiency significantly. This valve being slamproof and chatter-free is particularly pertinent to this station where the design makes use of a minimum-strength penstock. The construction of the valve also lends itself to maintenance economy by its seat arrangement and provision for seat adjustment and repairs. A commendable feature of the valve is its extremely low leakage characteristic. This feature would be highly advantageous on a reaction unit installation where watertight wicket-gate closure is more difficult to attain than watertight power-needle closure.

Power-Jet Control. The power-jet control has many desirable features embodied on its separate needle and power-jet-control operation.

The deflector-positioner cam which causes the deflector to follow the jet diameter eliminates dead-band difficulties permitting fast and uniform regulation characteristics throughout the entire load range.

The 2-sec full-load-rejection characteristic is instrumental in effecting lower penstock costs by minimizing penstock pressure rises, consistent with good regulation. This type of power-jet control would be even more advantageous for regulating performance on a unit designed for faster power-needle control. Individually controlled power needles provide for flexibility and most efficient operation over the complete load range.

Authors' Closure

The authors appreciate the comments by G. R. Woodman in which he approves of many of the novel design features of the turbine.

With reference to the over-all economy of draining the Villa Victoria Reservoir within a period of 6 months in order to save evaporation losses, this is only feasible because the flow from other reservoirs and other sources is more than sufficient to operate the downstream plants during the rainy season. Therefore the water from the Villa Victoria Reservoir could not be used by the downstream plants during this period and would be wasted. Thus there are two reasons for operating the Villa Victoria Reservoir as proposed. One is to save evaporation and the other is to store water during the rainy season for use not only by the San Bartolo Plants, but also for use by the downstream plants during the dry season. The value of having the Villa Victoria Reservoir stored water available for the downstream plants during the dry season more than compensates for the cost of the greater installed capacity of the Bartolo Plants necessary for intermittent operation. The reduction of evaporation is an added saving.

Since the normal tail water at the San Bartolo Plant is one meter (3.3 ft) below the impulse-turbine runner, the loss in head compared to a Francis runner is 0.8 per cent. Since field tests on vertical six-jet impulse turbines have shown around 92 per cent efficiency, it is doubtful whether a Francis unit, with high leakage losses through the runner clearances under the high heads, would show any better efficiency. Thus the difference in plant efficiency between the two types of units is represented by the loss in head for the impulse turbine. This loss is more than compensated for by the higher part-load efficiencies of the impulse turbine, its simplicity of construction, and lower maintenance costs as demonstrated by actual experience with the two types of turbines under similar heads.

Mechanism of Cavitation Inception and the Related Scale-Effects Problem

By R. W. KERMEEN,[1] J. T. McGRAW,[2] AND B. R. PARKIN,[3] PASADENA, CALIF.

An experimental investigation of incipient cavitation is described and the relationship between these experiments and current theories is discussed. Experiments have indicated that the cavitation number for inception varies with free-stream velocity and body size for both streamlined and bluff bodies. Preliminary studies of the mechanism of cavitation were conducted in order to understand the reasons for the variations in cavitation performance. Photographic investigations have disclosed the effect of the boundary layer on cavitation inception for both streamlined and bluff bodies. In addition, this study has shown that tensions exist in the flow of ordinary water at incipient cavitation.

Introduction

THE majority of hydrodynamic problems which arise in the design of devices that are submerged in flowing liquids may be treated by means of the classical aerodynamic techniques for incompressible fluids. The application of such methods to the design of ship and torpedo propulsive systems, and for the experimental determination of force coefficients for underwater missiles, is well known. However, a feature which is peculiar to the flow of liquids is the phenomenon of cavitation; that is, when the static pressure at any point in a liquid flow is low enough, boiling occurs and a two-phase flow system results. Aside from its adverse effect on hydrodynamic performance, cavitation is often undesirable because of the noise and physical damage which it produces. The present-day emphasis on the design of machinery with better performance at higher speeds accentuates these problems.

Thus it is clear that cavitating flow is of increasing technical importance, for today the designer is confronted with the cavitation problem more often than has been the case in the past. As with single-phase liquid flows, specific design problems usually are too difficult to permit a theoretical solution even when the basic knowledge is at hand, so the required design information must be obtained experimentally. Furthermore, the prototype is often a large machine and full-scale information is too expensive or difficult to obtain. Consequently, the desired experiments are made on small models. Such test results are useful to the designer only if they can be interpreted in the light of known laws of similitude. Although much is known concerning the modeling of single-phase flow, the onset of cavitation involves properties of the liquid not previously considered. For example, such factors as vapor pressure, surface tension, dissolved-gas content and surface conditions may affect the inception of cavitation.

In order to learn more about the physical laws which govern the scaling of cavitating flows, an experimental program was initiated in the High Speed Water Tunnel (1)[4] at the California Institute of Technology. This paper presents a summary of the progress to date and indicates the direction of the most recent researches.

Background

Until Thoma (2) and others considered the vapor pressure p_v, cavitation was felt to depend on the dimensionless parameter $P/(\rho V^2/2)$ where P and V are the pressure and velocity characteristic of the system and ρ is the liquid density. The inclusion of the vapor pressure changed the dimensionless parameter to $K = (P - p_v)/(\rho V^2/2)$, which is known as the cavitation number. For full cavity flow, which is characterized by a single large vapor or gas cavity enveloping some portion of the body, Reichardt (3) has shown that the cavitation number is the essential quantity for determining the geometry of the flow past a given body. Of course for model experiments made with hydraulic machinery, the degree of cavitation determined by a change of performance gave K-values that were a useful indication for the prototype.

On the other hand, differences in the cavitation number for the onset or inception of cavitation on a model and on a full-scale ship propeller were reported by Gutsche (4) in 1932. He ascribed this difference entirely to Reynolds number effects. In contrast, Ward and Land (5) chose to present their experimental results for cavitation inception with no attempt at a cavitation-number correlation. Their conservatism was probably due to the general lack of knowledge concerning any scaling laws for incipient cavitation which existed when their investigations were reported in 1940. However, they suggested that the speed for cavitation inception conceivably could depend upon the size of the model and they recommended that additional work be done to investigate this question. More recently, Birkhoff (6) and others have pointed out that the modeling of cavitation also could depend upon the dissolved-air content in the liquid and that, perhaps, the nucleus (unwetted impurities) content, surface tension, and the thermodynamic parameter which affects bubble growth from the nuclei should be considered.

This historical sketch, although incomplete, clearly shows the scarcity of factual knowledge which was available when this investigation was initiated. The possibility that many parameters could bear on the scale-effects problem is also evident. The object of this paper is to indicate present progress and to show how this research has emphasized the lack of basic knowledge on the mechanism of cavitation inception, which is the subject currently being investigated.

Initial Theoretical Considerations

In view of the uncertain state of knowledge just outlined, the first phase of this investigation required a critical examination of

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, October 26, 1953. Paper No. 54—Mex-1.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

the current opinions on the mechanics of cavitation inception. Of the many ideas considered, the viewpoint of Plesset (7) appeared to offer the best framework within which progress could be made.

It is assumed that there are very small nuclei containing air or water vapor, or both, which are stabilized on small solid particles in the liquid. Boiling or cavitation is initiated from such nuclei since their absence would mean that very large surface-tension forces must be overcome before cavitation can start. The slight tension under which cavitation normally occurs lends plausibility to arguments favoring the existence of the nuclei. Of course the small nuclei cannot be seen with the unaided eye and their presence is evidenced only as bubble growth when the flowing liquid has transported them into low-pressure regions. This growth must be governed by the laws of bubble dynamics. Hence this general view of cavitation inception allows the problem to be treated in two parts: (*a*) The hydrodynamics of the flow field determine the nature of the forces acting on a nucleus; and (*b*) this "forcing function" and the dynamics of cavitation bubbles determine the growth of the nucleus into visible cavitation.

Because of the dynamic aspect of cavitation inception, the first step in the systematic investigation of cavitation scale effects was an analysis to determine the conditions for dynamically similar bubble growths (8). Because of the stringent conditions obtained, it appeared that no useful scaling laws could be produced when the differential equation for bubble growth was subjected to a similarity analysis. These results did indicate that there is no reason for expecting the cavitation number to be the only significant parameter for defining the inception of cavitation in flows around geometrically similar bodies.

Exploratory Experiments

From the foregoing considerations it appeared that cavitation inception could depend upon other factors such as model size, free-stream velocity, surface tension, dissolved-air content, and the liquid viscosity. The similarity analysis gave us no information about the relative importance of these factors, and a series of experiments were planned to investigate this question. These preliminary water-tunnel experiments were performed with three geometrically similar airfoils to determine the effects of model size and free-stream velocity upon incipient cavitation.

The airfoils were symmetrical Joukowski profiles with a maximum thickness of 12 per cent of the chord. The chord lengths were 2, 4, and 8 in., and all models were designed to span the water-tunnel working section so that, except for a hydrostatic gradient, there were no spanwise pressure variations on the hydrofoil (9). It was found that the degree of cavitation for a given cavitation number varied with model size and tunnel velocity. These tests also showed the cavitation number for incipient cavitation K_i depended on scale and velocity. A plot of these experimental results is shown in Fig. 1 where the cavitation number for incipient cavitation K_i is plotted versus Reynolds number Re for the three different models. It will be noticed that the cavitation number for incipient cavitation increases with velocity and that the incipient cavitation number is not a function of Reynolds number only.

CIT Scale-Effect Experiments

The hydrofoil data showed considerable experimental scatter and could not supply strong enough proof for definite conclusions concerning the effect of model scale on cavitation inception. Further, it was found that very slight deviations in the model shape or test conditions produced large differences in cavitation performance and that two-dimensional models were difficult to fabricate with sufficient accuracy. In order to overcome these serious difficulties, a program was undertaken in which

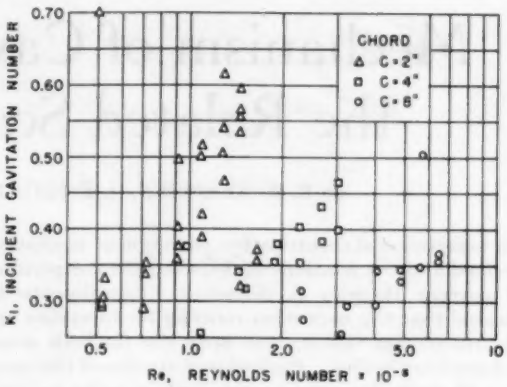

Fig. 1 Incipient Cavitation Number Versus Reynolds Number for Three Symmetrical Joukowski Hydrofoils

geometrically similar families of axially symmetric bodies were employed.

The first scale-effect experiments with axially symmetric bodies were performed on a family of cylinders with hemispherical noses (10). The model sizes ranged from $^1/_4$ to 2 in. diam and the free-stream velocity was varied between 10 and 100 fps. Although the primary purpose of these experiments was to investigate the variations of K_i with model scale and free-stream velocity, in some experiments the kinematic viscosity of the water was changed by altering the water temperature. These experiments demonstrated that cavitation inception definitely varied with both model size and free-stream velocity and, again, that the Reynolds number was not the primary parameter for defining incipient cavitation.

During these experiments it was found that the occurrence of cavitation, when the static pressure is reduced, may depend in a random manner upon the length of time during which the pressure is held at the low value. It is usually necessary to lower the pressure below the inception pressure before cavitation will begin. When cavitation first appears it is more fully developed than it is in the "incipient" state. Therefore it is necessary to raise the pressure to define the incipient condition. For brevity, these physical processes are called cavitation hysteresis. The term "incipient cavitation number," as used here, is defined as the state of liquid flow at which cavitation just disappears, or at which small wisps of cavitation are seen only intermittently as the static pressure is slowly increased at constant free-stream velocity.

This definition is only one of several ways by which the inception conditions could be determined. However, experience has shown that the definition adopted here gives more consistent data than other procedures. Further, it has the engineering advantage that conservative values for the incipient cavitation number are determined, and it simplifies the experiments by excluding any time dependence.

ORL-CIT Scale-Effect Experiments

The CIT experiments definitely indicated that cavitation inception varied with model size and tunnel velocity. However, all data had been obtained in the same facility and it was not known whether the results were affected by the characteristics of the specific tunnel. Since all the parameters influencing cavitation were not known, it was possible that the free-stream turbulence level or the presence or absence of a resorber might be important.

Accordingly, a joint experimental program was initiated with the Ordnance Research Laboratory at the Pennsylvania State University. The maximum model diameter could then be extended from the 2 in. of the CIT tests to 8 in. in the ORL's 48-in. Gar-

field Thomas Water Tunnel, and experiments on the same models in each facility could be compared in the overlapping range. Further, the ORL Tunnel did not have an air resorber (11) and was considerably different in arrangement and design from the CIT Tunnel.

This joint research program (12) further demonstrated the dependence of cavitation inception on model scale and water velocity. The close agreement of the results obtained from the two tunnels on the same set of hemispherical and 1.5-caliber ogive-nosed cylinders indicated that cavitation inception was not affected by the presence of a resorber on the CIT tunnel or by any other differences in the tunnel configuration, instrumentation, or in the properties of the water used. Faired curves from all data on the incipient cavitation characteristics of the axially symmetric bodies are shown in Fig. 2 where the incipient cavitation number is shown as a function of free-stream velocity. The incipient cavitation number increases with model size and free-stream velocity and approaches the magnitude of the minimum pressure coefficient on the body. Hence the usual procedure for predicting inception by equating the incipient cavitation number to the magnitude of the minimum pressure coefficient on the body gives conservative results.

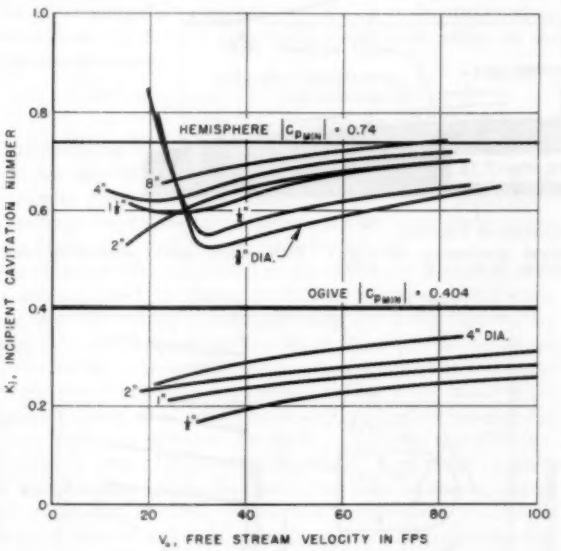

FIG. 2 INCIPIENT CAVITATION NUMBER VERSUS FREE-STREAM VELOCITY

(Upper set of curves is for bodies with hemispherical noses and lower set for 1.5-caliber ogive noses.)

Another important result which was obtained from all these experiments was that no consistent changes of the incipient cavitation number with dissolved-air content were observed. The air content in these experiments varied from approximately 7 to 13 ppm. No attempt was made to reconcile this finding with the contrary results of Numachi (13).

DISCREPANCIES BETWEEN EXPERIMENTAL AND THEORETICAL TRENDS

A theoretical investigation (14) which was conducted concurrently with the CIT scale-effect experiments gives only qualitative agreement with experimental trends. One of the major simplifying assumptions of the theory was the hypothesis that the bubble nuclei moved over the body with the velocity of the liquid outside the boundary layer. This severe restriction was imposed for two reasons. First, there was no knowledge of any boundary-layer effects, and the neglect of any possible viscous effects greatly simplified the calculations. Second, Knapp and Hollander's (15) observations of cavitation bubbles made this simplification seem realistic. The systematic variations between the calculated and experimental values of the incipient cavitation number made it evident that the neglect of the boundary layer was not justified. These differences between experiment and theory clearly indicated the need for detailed information on the mechanics of cavitation inception and the effect of the boundary layer on this process.

INCEPTION MECHANISM ON STREAMLINED BODIES

During the scale-effect experiments (10) small glistening bubbles were observed just ahead of the fine line of incipient cavitation. These bubbles were so small (0.001-in. to 0.002-in. diam) that they were visible to the unaided eye only when the lighting was so arranged that reflected high lights from the bubbles were intensified. The first photograph of these small bubbles, together with the commonly visible band of incipient cavitation, is shown in Fig. 3. This photograph was taken on a 2-in-diam black anodized dural hemispherical nose. For brevity, the band of cavitation, which is always visible to the unaided eye, will be called "macroscopic cavitation," while the very small bubbles which appear upstream of the macroscopic cavitation will be called "microscopic cavitation." The first observations of microscopic cavitation provided a definite indication that the assumption, that the nuclei traveled with the flow outside the boundary layer, may have been contrary to fact and that the resulting calculations did not represent the true physical situation. Accordingly, an experimental program for investigating the interaction of incipient cavitation and the boundary layer on smooth bodies was carried out (16).

Magnified double-flash still photographs were obtained with a plan view of the cavitation. From a knowledge of the flash rate of the photos it was possible to examine the multiple images and calculate the average velocity of the bubbles. It was found that

FIG. 3 INCIPIENT CAVITATION ON A 2-IN-DIAM BLACK ANODIZED HEMISPHERICAL NOSE

(Flow direction is from right to left. Small specks at right of larger white cavitation patches are termed microscopic cavitation.)

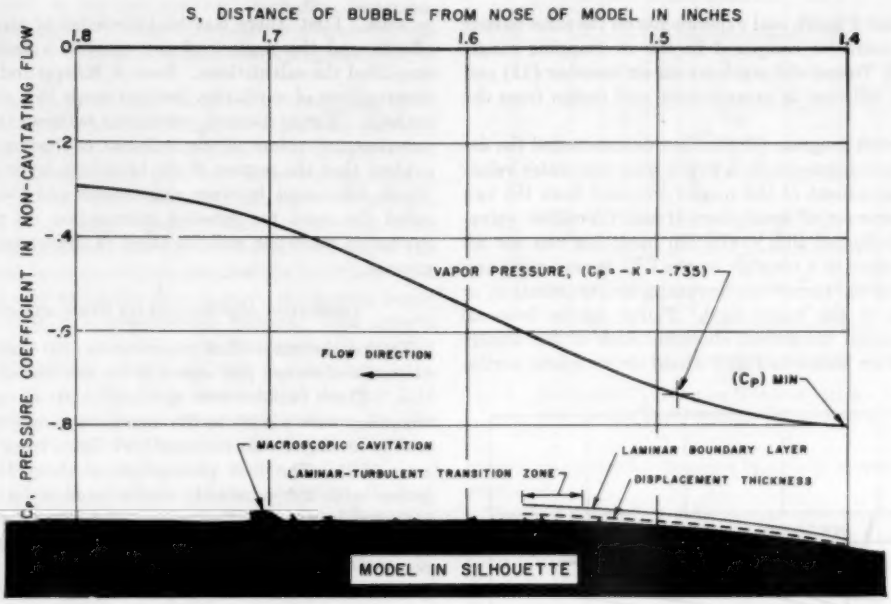

FIG. 4 INCIPIENT CAVITATION IN PROFILE

(Microscopic cavitation is shown in boundary layer upstream of zone marked macroscopic cavitation. Point where static pressure on body equals water-vapor pressure is denoted by $C_p = -K = -0.735$.)

the average velocity of the bubbles was much less than the free-stream velocity. This result indicated that the bubbles must be immersed in the boundary layer.

A second series of flash photographs were taken to examine a profile view of the cavitation to determine the location of the bubbles with regard to the model surface. A typical photograph from this series is shown in Fig. 4. This silhouette was taken at a cavitation number of 0.735 at a magnification ratio of 8. In this photograph the tiny bubbles inside the boundary layer can be seen upstream from the macroscopic cavitation. The boundary-layer thicknesses for noncavitating flow, which are superposed on the photographs, were calculated by the method of C. B. Millikan (17). The pressure-distribution coefficients C_p, added to the figure, were obtained from noncavitating flow measurements.

The point of cavitation inception is commonly recognized by the appearance of the foamy patch of macroscopic cavitation as indicated in Fig. 4. In addition to the macroscopic cavitation, this picture also shows small microscopic cavitation bubbles upstream of the macroscopic cavitation. The largest of these small bubbles in the boundary layer is little more than 0.001 in. diam.

The still photographs indicated that microscopic bubbles grow in the boundary layer and travel with less than free-stream velocity. In order to determine the relation between the growth of the microscopic bubbles and the macroscopic cavitation, it was necessary to obtain a more detailed history of the bubble motions. This was done with high-speed photography at 6000 frames per second. The time versus travel along the model of some typical small bubbles, as recorded by the motion pictures, is shown in Fig. 5. The motion pictures show that the bubbles grow for a time at a fixed position on the body. Then, when they have attained diameters roughly equal to the displacement thickness, they are lifted up slightly from the body, transported downstream, and distorted into irregular patches. The transport velocities shown by the graph ranged between 13 and 26 fps. The average of these bubble-transport velocities was 21 fps, compared to the 40 fps velocity of the oncoming flow.

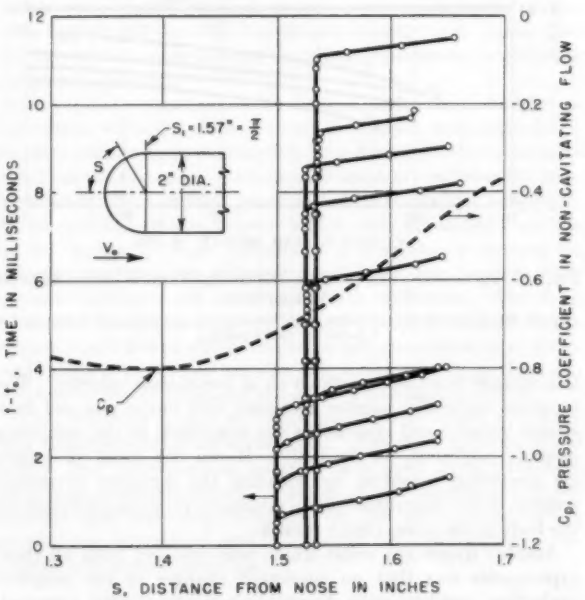

FIG. 5 MICROSCOPIC BUBBLE POSITION VERSUS TIME FOR A NUMBER OF TYPICAL BUBBLES

The mechanism of cavitation appears to change as the cavitation number is lowered. In the silhouette of Fig. 6 the cavitation number was lowered to a value which allows the foamy patch to become a series of attached clear cavities. The actual length of the cavity shown in the photograph was about $\frac{1}{4}$ inch. Characteristic of such attached cavities is suppression of the microscopic bubble growth. Apparently these cavities are sustained by evaporation and air diffusion through the cavity wall, as is the case for full cavity flow (3).

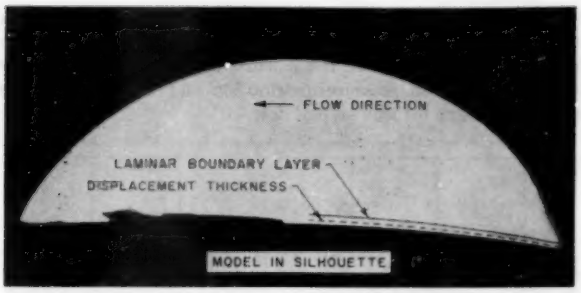

FIG. 6 CAVITATION IN PROFILE
(This is the same body shown in Fig. 4, with cavitation number K decreased from 0.735 to 0.698. Macroscopic cavitation has become a clear cavity with its leading edge attached to the body. Microscopic cavitation has been suppressed.)

As in the scale-effect experiments, a change in the dissolved-air content from 7.6 to 16.3 ppm had no detectable effect on the cavitation processes.

MINIMUM PRESSURE MEASUREMENTS AT CAVITATION INCEPTION

The photographic study made it possible to obtain information about the motion of cavitation bubbles in the boundary layer. In order to make theoretical calculations for bubble growth it is necessary to have information concerning the magnitude as well as the duration of the forces acting on the bubble.

Experiments were conducted to determine the minimum pressure existing on a body during cavitating conditions. Such work, using taps for pressure measurements, has been reported by Rouse (18) although any difficulties in these measurements are not mentioned. The present authors have found that premature cavitation in the pressure taps caused boiling of the water in the manometer lines, so that the manometer reading corresponded to the vapor pressure of water. In order to overcome this difficulty, experiments were conducted with pressure taps 0.006 in. diam and with silicone fluid in the manometer lines. Although reductions in tap diameter reduced the free-stream static pressure at which the taps cavitated, it was not possible to suppress the premature cavitation. Even after the silicone fluid had been heated to drive out the dissolved air, premature bubbling of the silicone in the tap occurred.

Since conventional techniques could not provide direct indications of the minimum pressure on a body with incipient cavitation, a new method for measuring the pressure was developed. A thin rubber membrane was cemented over the entire model nose. The membrane covered the single $^1/_{16}$-in-diam tap at the minimum-pressure point. When the nose was installed in the tunnel working section, a vacuum pump and mercury manometer with a reservoir were connected to the tap as shown in Fig. 7. The mercury reservoir on the manometer could be raised or lowered to change the air pressure in the tap and to cause the membrane to bulge in or out. A cathetometer telescope was placed near the working-section window so that the deflection of the membrane could be observed. By adjusting the lighting of the membrane, the telescope cross hair could be set so that very small changes of light and shadow on the membrane permitted accurate determination of the membrane null position. When the membrane was nulled, the mercury manometer gave a direct measure of the pressure in the liquid flow outside of the membrane.

Of course at incipient cavitation where actual liquid tensions existed, the membrane was observed to bulge slightly, and for fully developed cavitation the membrane could not always be seen. This uncertainty in the measurements was partly overcome by calibrating for membrane bulge and subtracting this

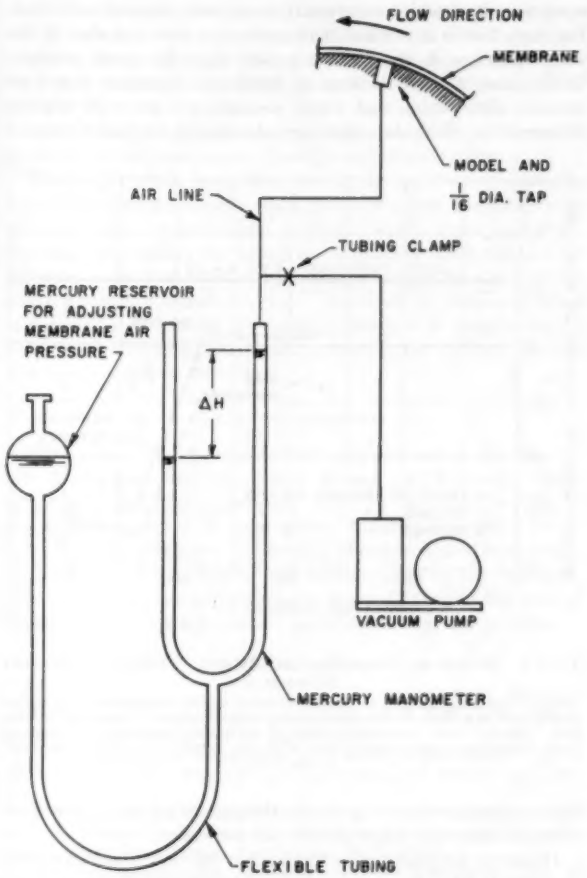

FIG. 7 SCHEMATIC DIAGRAM OF APPARATUS FOR MEASURING MINIMUM PRESSURE

calibration pressure from the manometer readings. The data were corrected in such a way that the true tensions in the water were greater than or equal to the measured values. Instrumentation for overcoming this difficulty is now being developed.

An interesting qualitative result was provided by this first pressure cell. As the free-stream pressure was lowered from the incipient state to obtain more fully developed cavitation, the slightly bulging membrane was always suddenly pushed down into the hole by the increase in local pressure when the macroscopic cavitation first became a small, clear cavity attached to the model downstream from the tap. Thus the upstream influence of the clear macroscopic cavitation was shown to be significant.

The results of the experiments are shown in Fig. 8 in which the minimum absolute pressure on the model is plotted against the cavitation number. It will be noted that in the incipient cavitation-zone pressures were measured which were below vapor pressure. In fact, negative pressures or tensions were measured. It is interesting to note that the tensions given by Fig. 8 correspond to approximately 4 deg F superheat for boiling, which is nearly the same superheat found by Dergarabedian (19) in his boiling experiments, and correspond roughly to Plesset's assumption for tension in his theoretical analysis of cavitation-bubble growth (7).

ROLE OF BOUNDARY LAYER IN BLUFF-BODY CAVITATION

So far we have considered the effects of the boundary layer

upon cavitation which occurs only on smooth, streamlined bodies. For such bodies it is clear that cavitation does not start if the lowest pressure on the body is greater than the vapor pressure. This means that predictions of cavitation inception based on pressure distribution and vapor pressure are generally slightly conservative, since the experimental value of cavitation number

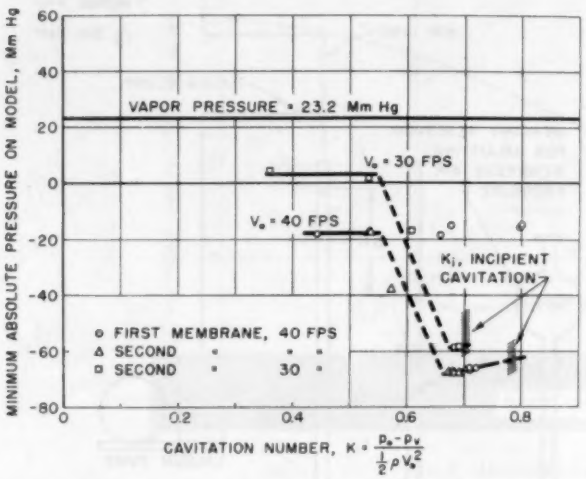

FIG. 8 MINIMUM PRESSURE ON MODEL VERSUS CAVITATION NUMBER K

(Dotted portions of curves denote regions where macroscopic cavitation first becomes a clear cavity and thereby suppresses the microscopic cavitation. Shaded areas represent region of cavitation inception. At right of these zones there was no cavitation, while the extent of cavitation increased to the left.)

for inception seems to approach the calculated value as a limit as model scale and water velocity are increased.

However, for bluff bodies which allow the flow to separate from the solid surface, the simple relationship between the liquid vapor pressure and the pressures on the body need not apply. Instead, the low pressures required for cavitation-bubble growth could be produced by the vorticity in the wake with the first evidences of cavitation centered in the vortex cores. Since the intensity of the wake vorticity is approximately determined by the vorticity in the boundary layer at the separation point, we may regard incipient cavitation in the wake of a bluff body as another example of the influence of the boundary layer on cavitation inception.

CAVITATION INCEPTION FOR BLUFF BODIES

In order to determine the effect of model size and velocity on cavitation inception for bluff bodies, a series of water-tunnel experiments were planned, utilizing a family of sharp-edged circular disks ranging in size from $^1/_{16}$ to $1^1/_2$ in. diam. Disks were chosen for simplicity of fabrication and because flow separation occurs at a well-defined point.

The experiments (20) which were made on the family of disks at various free-stream velocities and at two temperatures, 55 deg and 77 deg F, showed that the cavitation number K_i for inception in the wake, varied with model size and free-stream velocity as had been the case for streamline bodies. Cavitation hysteresis was again observed so the same test procedure was used to obtain the data. These experiments indicated that for this particular flow, the Reynolds number based on disk diameter was the primary scaling parameter. The experimental cavitation number K_i, plotted as a function of Reynolds number, is shown in Fig. 9. The minimum-pressure coefficient, based on the lowest measured pressure on the disks, is plotted in the same figure. Incipient cavitation behind the disks occurs at much higher cavitation numbers than would be indicated from these lowest pressure measurements. Therefore the calculated cavitation number for inception based on pressure distribution and vapor pressure is not conservative for this type of flow.

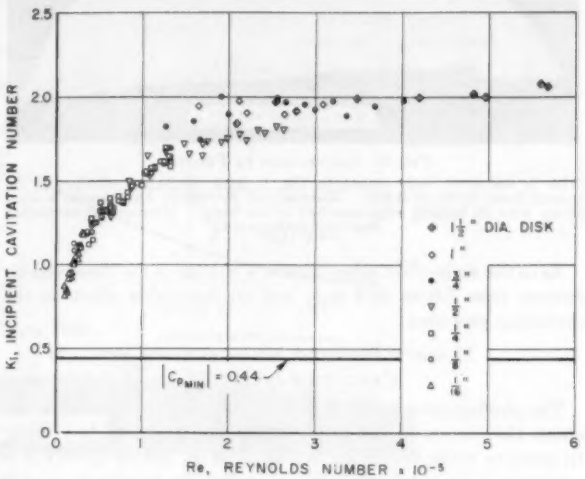

FIG. 9 INCIPIENT CAVITATION NUMBER K_i VERSUS REYNOLDS NUMBER BASED ON DISK DIAMETER

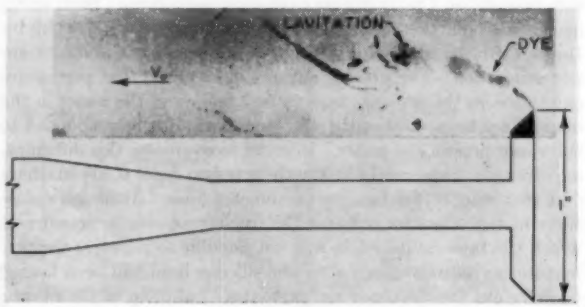

FIG. 10 INCIPIENT CAVITATION BEHIND A SHARP-EDGED DISK

(Dye has been injected into boundary layer on front surface of disk to show how cavitation is associated with wake flow. It can be seen that liquid from disk boundary layer rolls up into vortex cores from which cavitation originates.)

Fig. 10 shows how the first signs of cavitation occur in the wake. Owing to the shedding of vortexes, high rotational velocities occur in the flow which are accompanied by correspondingly low local pressures at the vortex cores which lead to cavitation. An example of the rotational speeds in a vortex core is provided by the series of photographs in Fig. 11. These magnified pictures are taken from the 6000-frame-per-sec motion pictures and show that the two small bubbles of approximately 0.002-in. maximum diameter have a relative angular velocity of about 30,000 rpm. Such very high rotational speeds give a convincing explanation for the cavitation in the wake despite the high ambient pressures.

CONCLUSIONS

The work outlined in this paper is in no sense complete; however, definite results on the questions of the mechanism of cavitation inception and the related scale-effects problem have been found.

For example, the experimental programs at CIT and ORL have established the variation of incipient cavitation number with free-stream velocity and model scale. For smooth, streamlined

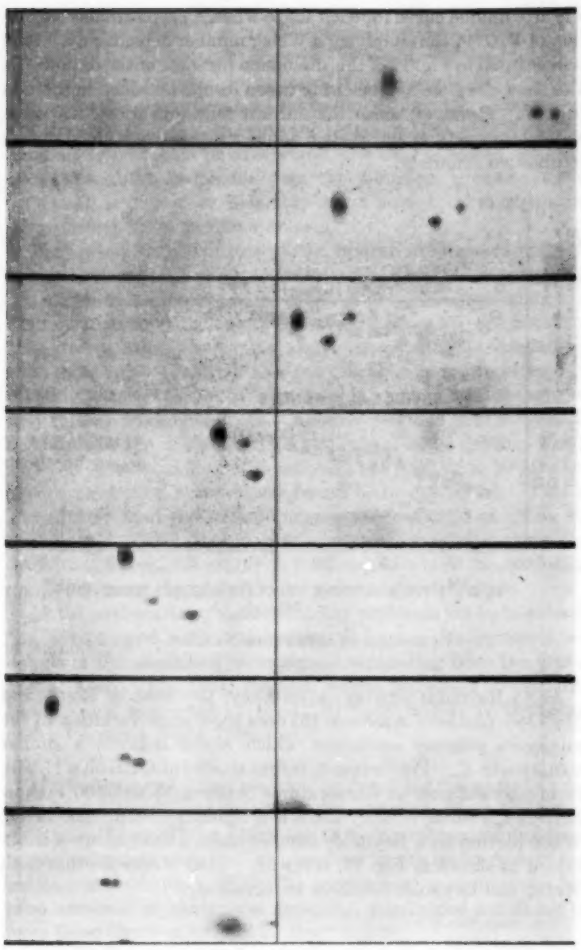

FIG. 11 CAVITATION BUBBLES IN A ROTATING VORTEX CORE
(These magnified pictures are from a motion-picture sequence which was taken at 6000 frames per sec in the shear layer immediately behind a sharp-edged disk.)

bodies the incipient cavitation number increases with model size and free-stream velocity. As scale and velocity increase, the incipient cavitation number approaches the magnitude of the minimum pressure coefficient on the body. Therefore the usual procedure for predicting the incipient cavitation number by equating it to the magnitude of the minimum-pressure coefficient gives conservative results.

The experimental trends for the variation of the incipient cavitation number with free-stream velocity and model size for the sharp-edged disks showed that the Reynolds number is a significant scaling parameter. The minimum-pressure coefficient on the body could not be compared with the incipient cavitation number because the first signs of cavitation are centered in the wake-vortex cores. Within the range of the present disk experiments, the incipient cavitation number did not approach any limiting value as the Reynolds number increased and the incipient cavitation number was always greater than the magnitude of the minimum-pressure coefficient on the body.

The photographic investigations of the effect of the boundary layer on cavitation inception for smooth streamlined bodies and for bluff bodies which have been completed gave significant new information concerning the mechanics of cavitation inception. For streamlined bodies it has been found that cavitation bubbles originate from very small bubbles in the boundary layer, while for the sharp-edged disks the first signs of cavitation arise from very small bubbles which grow in the vortex cores of the wake.

For the first time, tensions have been observed and measured in the flow of ordinary water during incipient cavitation.

The results which have been found thus far are not extensive enough to allow a complete analysis of cavitation inception to be made. There are still many problems which must be solved. For example, some of the future investigations must consider air diffusion, bubble dynamics, and the hydrodynamics of bubbles which are in the boundary layer. Research on these and other important aspects of the cavitation problem is in progress in the hope that a complete theoretical explanation for the experimental observations will be developed.

ACKNOWLEDGMENT

The authors wish to express their appreciation to the Office of Naval Research and the Navy Bureau of Ordnance, which sponsored these investigations. We are grateful to Dr. J. Robertson and Mr. J. W. Holl of the Ordnance Research Laboratory, Pennsylvania State University, who co-operated in a portion of the work. We have also been greatly aided by Dr. Milton S. Plesset of the Hydrodynamics Laboratory, California Institute of Technology, who has furnished guidance during this program.

BIBLIOGRAPHY

1 "The Hydrodynamics Laboratory of the California Institute of Technology," by R. T. Knapp, J. Levy, J. P. O'Neill, and F. B. Brown, Trans. ASME, vol. 70, 1948, p. 437.

2 "Experimental Research in the Field of Water Power," by D. Thoma, Trans. First World Power Conference, vol. 2, 1924, pp. 536–551.

3 "Reports and Translations No. 776," by H. Reichardt, Ministry of Aircraft Production; distributed by the Office of Naval Research, Navy Department, Washington, D. C., 1946.

4 "Hydromechanishe Probleme des Schiffsantriebs," by F. Gutsche, editors—G. Kempf and E. Foerster, printed by Freunde and Forderer, Hamburg, Germany, for the Hamburg Towing Tank, 1932.

5 "Preliminary Tests in the NACA Tank to Investigate the Fundamental Characteristics of Hydrofoils," by K. E. Ward and N. S. Land, NACA ARC (WR L 766), September, 1940.

6 "Hydrodynamics," by G. Birkhoff, Princeton University Press, 1950; see Sect. 16, p. 106. References to the work of the original investigators are given there.

7 "The Dynamics of Cavitation Bubbles," by M. S. Plesset, *Journal of Applied Mechanics*, Trans. ASME, vol. 71, 1949, pp. 277–282.

8 "Scale Effects in Cavitating Flow—A Preliminary Report," by B. R. Parkin, California Institute of Technology, Hydrodynamics Laboratory Report No. 21-7, December, 1951.

9 "The Effect of Wall Interference Upon the Aerodynamic Characteristics of an Airfoil Spanning a Closed-Throat Circular Wind Tunnel," by W. G. Vincenti and D. J. Graham, NACA TR 849, 1946.

10 "Some Observations of Cavitation on Hemispherical Head Models," by R. W. Kermeen, California Institute of Technology, Hydrodynamics Laboratory Report No. E-35.1, June, 1952.

11 "Air Resorption in Water Tunnels," by F. B. Brown, California Institute of Technology, Hydrodynamics Laboratory Report No. N-62, March, 1949.

12 "Incipient-Cavitation Scaling Experiments for Hemispherical and 1.5-Caliber Ogive-Nosed Bodies," by B. R. Parkin and J. W. Holl; a joint study by the Hydrodynamics Laboratory, California Institute of Technology, and Ordnance Research Laboratory, The Pennsylvania State University, May, 1953.

13 "The Effect of Air Content on the Appearance of Cavitation in Distilled, Salt, and Sea Water," by F. Numachi and T. Kurokawa, Translation and Commentary, Ordnance Research Laboratory, The Pennsylvania State University, August, 1946.

14 "Scale Effects in Cavitating Flow," by B. R. Parkin, California Institute of Technology, Hydrodynamics Laboratory Report No. 21-8, July, 1952.

15 "Laboratory Investigations of the Mechanism of Cavitation," by R. T. Knapp and A. Hollander, Trans. ASME, vol. 70, 1948, pp. 419–435.

16 "Incipient Cavitation and Boundary Layer Interaction on a Streamlined Body," by B. R. Parkin and R. W. Kermeen, California Institute of Technology, Hydrodynamics Laboratory Report No. E-35.2, December, 1953.

17 "The Boundary Layer and Skin Friction for a Figure of Revolution," by C. B. Millikan, Trans. ASME, vol. 54, Paper APM—54-3, 1932.

18 "Cavitation and Pressure Distribution—Head Forms at Zero Angle of Yaw," by H. Rouse and J. S. McNown, Bulletin 32, State University of Iowa, Studies in Engineering.

19 "The Rate of Growth of Vapor Bubbles in Supersaturated Water," by Paul Dergarabedian, California Institute of Technology, Hydrodynamics Laboratory Report No. 21-10, August, 1952.

20 "Incipient Cavitation and Wake Flow Behind Sharp Edged Disks," by R. W. Kermeen and B. R. Parkin, California Institute of Technology, Hydrodynamics Laboratory Report No. E-35.3, January, 1954.

Discussion

J. W. Holl[5] and J. M. Robertson.[6] This welcome paper on a subject of vital interest to everyone involved in the prediction of the occurrence of cavitation on bodies is of special interest to the writers because of the joint experimental program referred to. Our interest stems from the fact that the Ordnance Research Laboratory's 48-in. water tunnel was built by the Navy Bureau of Ordnance primarily for experimental cavitation studies on 5-in. propellers simulating propellers some 15 in. diam. Thus our primary interest is in the establishment of scaling laws for hydrofoils. The data presented in Fig. 1 of the paper indicate a fairly rapid decrease in K_i with increase in size for hydrofoils, while Fig. 2 for noses indicates a slight increase at the same velocity or Reynolds number. Research on hydrofoil cavitation is in progress in several tunnels to clarify this apparent anomaly.

The authors describe their visual method for determining the inception of cavitation. As they indicate, this is a good engineering procedure from the standpoint of avoiding the uncertainties of the hysteresis phenomena. The writers wish to note, however, that there are several criteria for defining a critical cavitation number. If one is interested in the production of noise, then acoustic means must be employed to determine the inception of cavitation. The incipient cavitation number determined by such a method is generally greater than that determined by visual means. The agreement between the two values depends primarily upon how well the test body may be visualized. When one is not interested in noise but rather the effects of cavitation on the hydrodynamic forces, then the critical number is defined as the point where the lift or drag first starts to change its characteristics with decreasing cavitation number. This critical value tends to be less than that determined visually. Thus the method used by the authors is seen to be reasonably satisfactory.

The authors state that no useful scaling laws could be deduced from the differential equations describing the bubble growth. However, using the dimensional-analysis approach and employing the measurable variables of model size, free-stream velocity, surface tension, dissolved-air content, and liquid viscosity and density, one can obtain possible scaling parameters. The result of such an analysis indicates that K_i should be a function of the air content, the Reynolds number Re, and the Weber number $W(W = V_0 D^{1/2}/(s/\rho)^{1/2}$, where D is the characteristic length, s the surface tension, and ρ the density.

As shown in reference (12) of the paper, the data of Fig. 2 yield a nearly unique curve for each shape when K_i is plotted as a function of $V_0 D^{1/2}$, thus implying a Weber number dependency. This is illustrated in Fig. 12 of this discussion for some of the data which also show the close agreement between results obtained in the two tunnels. However, since the surface tension varies little with water temperature, the Weber number dependency could not be established definitely.

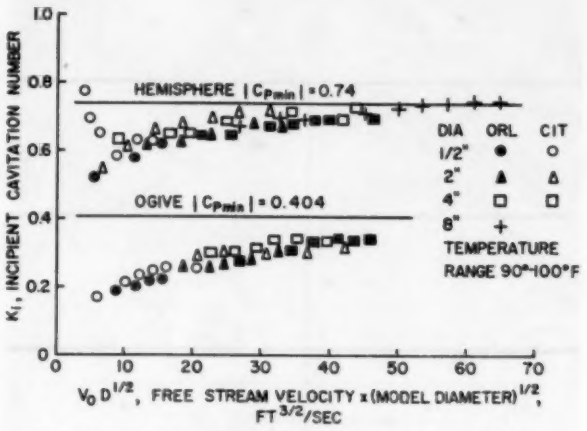

Fig. 12 Variation in Inception Number With $V_0 D^{1/2}$

As to Reynolds number dependency, the work of Rouse and McNown (authors' reference 18) does show some variation in the minimum pressure coefficient which would indicate a similar variation in K_i. Furthermore, recent studies at ORL on a $1^1/_2$-in. hemispherical nose at temperatures between 55 and 100 F show that at the same velocity there is a significant difference in K_i. When plotted on a Reynolds number basis, a unique curve is obtained as shown in Fig. 13, herewith. Thus it appears that both Weber and Reynolds numbers are involved.

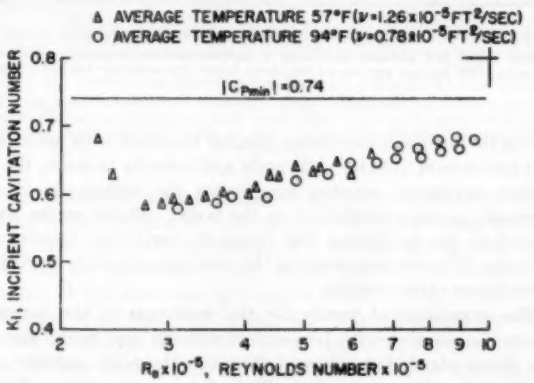
Δ AVERAGE TEMPERATURE 57°F (ν = 1.26 x 10^{-5} FT^2/SEC)
○ AVERAGE TEMPERATURE 94°F (ν = 0.78 x 10^{-5} FT^2/SEC)
| C_{Pmin} | = 0.74
K_i, INCIPIENT CAVITATION NUMBER
R_e x 10^{-6}, REYNOLDS NUMBER x 10^{-5}

Fig. 13 Variation in Inception Number With Reynolds Number for $1^1/_8$ In. Hemispherical Nose

As the authors note, cavitation-inception tests on nose shapes have not indicated any significant variation with air content. However, we are aware of tests of other objects, such as propellers, in which variations are apparently due to air content.

The presentation of the information on the inception mechanism on streamlined bodies is the first to show the fine details of bubble growth on a surface. From the further work promised by the authors we look forward to obtaining an even clearer picture

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[6] Professor of Engineering Research and Assistant Director, Ordnance Research Laboratory, The Pennsylvania State University. Mem. ASME.

of the phenomenon. A most novel technique is presented for the measurement of negative pressures. The authors' description of the difficulties involved in pressure measurements by the usual techniques for conditions near cavitation explain the anomalies of the Rouse-McNown data, authors' reference (18). Until now we have never been able to understand why they obtained incipient cavitation numbers greater than the minimum pressure coefficient. It is now clear that they must have been experiencing cavitation in the piezometer holes.

The writers also have studied the cavitation inception on bluff bodies although not so extensively as the Cal Tech researchers. Thus for a 4-in. disk, at a speed of about 30 fps (corresponding to a Reynolds number of about 10^6) a value of $K_i = 2.7$ was obtained acoustically. This is somewhat above the trend of the data in Fig. 9 of the paper. Whether this is explicable in terms of experimental scatter or is due to differences in experimental technique, only further research will tell. Another common bluff body that has been studied in the ORL Water Tunnel is the sphere. For 2 and 4-in. spheres at Reynolds numbers of 7 and 12×10^5, the incipient cavitation number was found to be 1.6 to 2.0. This is also greater than the minimum pressure coefficient of about 1.2 (for $Re > 5 \times 10^5$). It is interesting to speculate on what the cavitation inception number is for flows in which the Reynolds number is below the critical for boundary-layer transition.

As the authors note, there are many problems yet to be solved. We look forward, with anticipation, to further disclosures of the details of the cavitation phenomenon emanating from the group at Cal Tech. The ORL research results presented in this discussion were supported by the U. S. Navy Bureau of Ordnance under Contract NOrd 7958.

M. S. Plesset.[7] The research reported by the authors constitutes a valuable contribution to the understanding of incipient cavitation flow. The authors are to be complimented for their ingenious measurements of the tensions in the water flow in the cavitating condition. While the existence of such tensions has been assumed on theoretical grounds,[8] it is indeed satisfying to have these direct experimental observations.

Attention may be called to the fact that, in these experiments, no relatively large bubbles were observed; nor were bubbles observed to collapse and rebound. It is now clear that so-called cavitation bubbles, for which rebound has been reported,[9] are air bubbles which pulsate in the pressure field in the neighborhood of a water-tunnel model. The observations of Ellis[10] may also be cited in this connection. Ellis observed that vapor bubbles collapsed with no rebound whatever. Air bubbles, on the other hand, would oscillate in size through many cycles as do underwater explosion bubbles.

The discovery reported in this paper that cavitation inception is associated with the appearance of microscopic bubbles in the boundary layer of the submerged body is clearly of great interest and importance. Many interesting lines for further investigation are now indicated. For example, the authors' Fig. 4 shows that the microscopic cavitation bubbles continue to grow in a boundary-layer region where the pressure has risen above vapor pressure. This result indicates that diffusion of dissolved air from the supersaturated water in this region is a decisive mechanism for the growth of these microscopic cavitation bubbles. Unfortunately, no adequate theory for the rate of diffusion under the given convection conditions is as yet available.

Irving Taylor.[11] In the next-to-last paragraph under Conclusions, the authors state, "For the first time, tensions have been observed and measured in the flow of ordinary water during incipient cavitation." Possibly the word ordinary should be emphasized more; otherwise the statement would seem to overlook the findings of S. F. Crump, published in 1949.[12]

Authors' Closure

With regard to the dependency of the incipient cavitation number on Reynolds number, the authors agree with Messrs. Holl and Robertson that the variations in minimum pressure coefficient with Reynolds number indicate a similar variation in K_i. This effect should be of importance only in the range below the critical Reynolds number which for hemispheres is approximately 2×10^5. The temperature range of 55 to 100 F reported in tests by Holl and Robertson far exceeds the variations obtained in the tests reported by the authors and, in fact, no systematic variations of temperature were made except in the tests on bluff bodies. Fig. 13 in the discussion clearly shows a Reynolds number dependency in tests on one model; however, the tests by the authors do not show this dependency for streamline bodies when Reynolds number is varied by changing model size. This is particularly true for the hydrofoils, Fig. 1. Thus there appears to be a Reynolds number dependency for any one model but no clear one between models of varying sizes.

The authors agree with the remarks of Holl and Robertson on the effects of air content on cavitation inception and find their observations about cavitation tests on propellers interesting. In a recent conversation at California Institute of Technology, Dr. H. Lerbs, of the David Taylor Model Basin, told us of his experiments which have led him to conclude that cavitation inception on ship propellers is little influenced by dissolved-air content. These findings contradict the previous results mentioned by Holl and Robertson. Certainly our understanding of cavitation inception will not be complete until we can explain such conflicting results.

Professor Plesset's comments as to the role of air diffusion in the growth of the microscopic cavitation bubbles are important. It is the belief of the authors that this growth by air diffusion occurs in the later stages of growth. Present evidence (16) indicates that at inception the microscopic cavitation originates in regions of the boundary layer where the static pressure is less than the liquid-vapor pressure. That is, initially, cavitation nuclei grow only by dynamic means, and air diffusion is important only after bubble growth has started.

With regard to the question raised by Mr. Taylor—the authors are familiar with the work of S. F. Crump, who, like F. Numachi (13) and others, has done important work on cavitation in venturi nozzles. Mr. Crump did not measure tensions in water but did predict their existence from measurements made in a noncavitating region of the flow circuit. In fact, Mr. Crump states,[12] "Direct measurement of P_s (critical pressure) at the constriction was purposely avoided for these tests as it was suspected that piezometer taps in the constriction might influence the inception of cavitation."

In conclusion, the authors would like to express their appreciation to those who commented on this paper.

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[7] Professor of Applied Mechanics, California Institute of Technology, Pasadena, Calif. Mem. ASME.

[8] "Dynamics of Cavitation Bubbles," by M. S. Plesset, *Journal of Applied Mechanics*, Trans. ASME, vol. 71, 1949, p. 277.

[9] "Laboratory Investigations of the Mechanics of Cavitation," by R. Knapp and A. Hollander, Trans. ASME, vol. 70, 1948, p. 419.

[10] A. T. Ellis, Hydrodynamics Laboratory Report, California Institute of Technology, No. 21-12 (1952).

[11] Head, Pump, and Compressor Section, The Lummus Company, New York, N. Y. Mem. ASME.

[12] "Determination of Critical Pressures for the Inception of Cavitation in Fresh and Sea Water as Influenced by Air Content of the Water," by S. F. Crump, David W. Taylor Model Basin Report No. 575, 1949.

Refrigerating-Plant Performance Characteristics by Electrical-Analog Analysis

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The purpose of this paper is to develop a means for predicting over-all performance characteristics of an integrated machine and heat-transfer complex. To this end electrical-analog methods are applied to the study of refrigerating-plant performance.

Nomenclature

The following nomenclature is used in the paper:

- A = heat-transfer surface area, sq ft
- C = clearance fraction; clearance volume versus compressor displacement
- c_b = evaporator-brine mean specific heat, Btu/(lb-deg F)
- c_c = condenser-coolant mean specific heat, Btu/(lb-deg F)
- E_v = compressor volumetric efficiency
- n = exponent in pv^n = const; ideally $n = k = c_p/c_v$, ratio of gas (vapor) specific heats
- p = pressure, psia
- Δp = pressure difference between evaporator and condenser, psi
- p_s = suction pressure of compressor, psia
- p_d = discharge pressure of compressor, psia
- q = heat energy as applied to different elements of system, Btu/hr, and identified by subscripts, such as in conjunction with compressor, i.e. "work," "evap," and "cond"
- R_c = condenser-coolant heat-exchange resistance, thohms, deg F/(Btu/hr)
- R_b = evaporator-brine heat-exchange resistance, thohms, deg F/(Btu/hr)
- $R_{t.c}$ = total equivalent thermal resistance for condenser, thohms, deg F/(Btu/hr)
- $R_{t.e}$ = total equivalent thermal resistance for evaporator, thohms, deg F/(Btu/hr)
- $R_{UA.c}$ = net condenser heat-transfer resistance, thohms, $= 1/(UA)_c$, deg F/(Btu/hr)
- $R_{UA.e}$ = net evaporator heat-transfer resistance, thohms, $= 1/(UA)_e$, deg F/(Btu/hr)
- r = electrical resistance corresponding to thermal resistance, ohms
- Δt = temperature difference, deg F, between evaporator and condensing temperature
- $\Delta t_{t.c}$ = temperature difference, deg F, between condensing vapor and coolant-inlet temperature
- $\Delta t_{t.e}$ = temperature difference, deg F, between evaporating vapor and brine-inlet temperature
- U = over-all heat-transfer conductance, Btu/(hr-sq ft-deg F), applicable, with subscript, to condenser or evaporator
- W_c = condenser-coolant flow rate, lb/hr
- W_b = evaporator-brine flow rate, lb/hr

Introduction

Prediction of the performance of a composite vapor-compression refrigeration plant proves difficult because of the many simultaneous variables in force at any one time. Whereas the design of a system may be made in terms of initially set conditions, still actually the operating conditions may be quite different from the original design. For example, a plant with a fixed compressor size designed to chill brine of a specified initial temperature and flow rate, under fixed condenser-coolant temperature and flow rate, actually may operate with brine of a different temperature. Under these circumstances, prediction of the performance of the composite system, as involving the plant equipment of fixed dimensions, may be quite difficult.

Investigation of performance under different operating conditions may be carried out through the extension of the resistance concept utilized in heat-transfer studies. Beyond this, the interdependent relationships for the system components may be investigated by electrical-analogy methods, wherein the performance values may be represented by electrical quantities. The application of the resistance concept and the use of electrical-analogy methods for the study of different steady-state heat-flow systems is well recognized (1, 2, 3, 4, 5),[2] and likewise so for unsteady-state systems (6). Extension of the resistance concept to include machine performance in conjunction with heat-transfer equipment has been studied on an introductory basis (7).

Reducing the refrigerating-plant components to those of primary significance, the following may be cited: (*a*) Vapor compressor, (*b*) refrigerant condenser, (*c*) refrigerant evaporator, (*d*) expansion valve (pressure-drop device) between the high and low side. Each of the first three components in particular has operating characteristics which depend not only on specific designs but also on the operating conditions to which they may be subjected. These operating conditions may be influenced mutually by the major components of the system. The general arrangement of the basic elements of the refrigerating system is shown in Fig. 1.

In the present analysis many simplifying assumptions will be found expedient. Idealized performance for the sake of clarification will be assumed, though corrections to account for performance of real equipment may be made readily. The essential object of the present paper is primarily for the purpose of developing further another tool for use in electrical-analogy operations. This tool may thus serve for prediction of over-all performance characteristics of an integrated machine and heat-transfer complex.

Vapor-Compression Refrigeration Circuit

In the conventional refrigeration system represented in Fig. 1, power is supplied to drive the compressor which pumps the refrigerant; the refrigerant is liquefied in the condenser through coolant circulation; brine, in turn, is chilled in the evaporator by

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, January 21, 1954. Paper No. 54—Mex-11.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

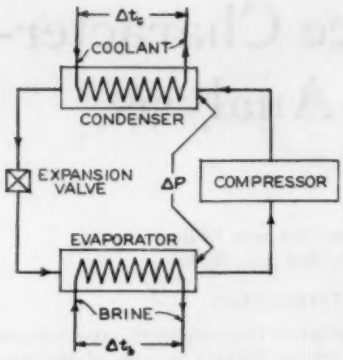

FIG. 1 BASIC ELEMENTS OF A VAPOR-COMPRESSION REFRIGERATION CIRCUIT

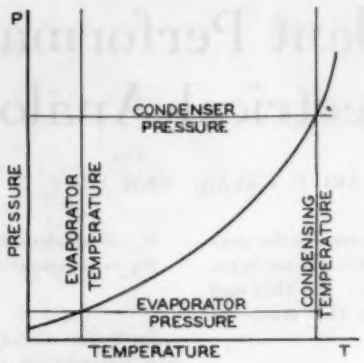

FIG. 2 TYPICAL PRESSURE-TEMPERATURE RELATIONSHIP FOR A REFRIGERANT

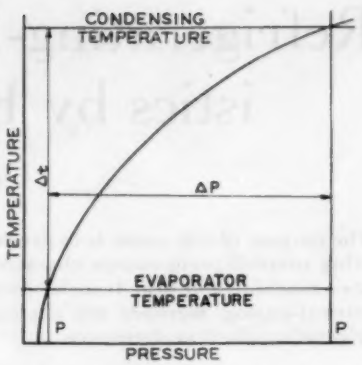

FIG. 3 TYPICAL TEMPERATURE-PRESSURE RELATIONSHIP FOR A REFRIGERANT

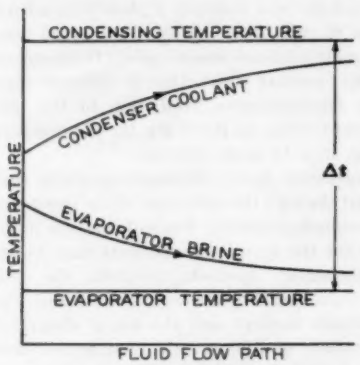

FIG. 4 RELATIVE SYSTEM HEAT TRANSFER-TEMPERATURE RELATIONSHIPS

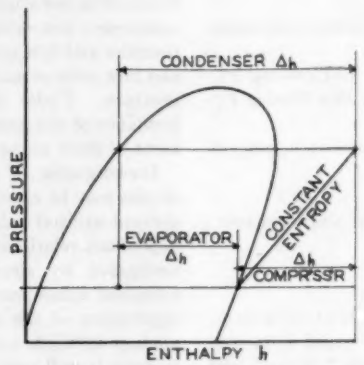

FIG. 5 TYPICAL PRESSURE-ENTHALPY (p-h) RELATIONSHIPS FOR VAPOR-REFRIGERATION CYCLE

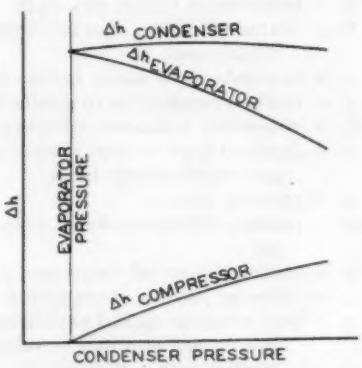

FIG. 6 RELATIVE VARIATIONS OF Δh FOR IDEAL CYCLE, FIXED EVAPORATOR PRESSURE, VARYING CONDENSING PRESSURE

the refrigerant which subsequently is recycled in the power-driven compressor. Fig. 2 shows the pressure-temperature relationship for a refrigerant, indicating the temperature-difference effect accomplished by the compressor in its generation of pressure difference. Temperature-pressure levels of the system are shown in Fig. 3 with the corresponding general relative temperature relationships for the heat-transfer equipment shown in Fig. 4.

Cycle Performance. In Fig. 5 the pressure-enthalpy (p-h) diagram for an ideal refrigeration cycle is shown, with particular reference to component enthalpy differences: i.e., Δh for the evaporator, the compressor machine, and the condenser. On the basis of a fixed evaporator pressure and varying condenser pressure, Fig. 6 shows the relative variations of Δh for the evaporator and the compressor, it being recognized that their sum represents the value of Δh for the condenser in the ideal cycle, as shown. (Liquid at the expansion valve is assumed at condensing temperature; saturated vapor to the compressor and isentropic compression in the machine are likewise assumed. The Δh-values of course represent the energy values per pound of refrigerant.)

Machine Performance. Although equivalent analysis of refrigerating-plant performance could be carried out with other forms of compression machines, in this paper a positive-displacement compressor with suction and discharge valves is assumed. Herein the effect of operating pressures and pressure difference, in conjunction with design details, on the pumping-capacity rate of the compressor is to be noted; that is, the ideal displacement gas-volume rate is modified by the effect of the over-all volumetric efficiency. The effective gas-handling capacity, as measured by real volumetric efficiency, is affected by the cylinder clearance volume in conjunction with the pressure ratio, valve pressure drops, suction gas heating, and the specific gas properties. The volumetric efficiency on an idealized cylinder basis, exclusive of valve drops and suction heating, may be cited

$$E_v = 1 + C - C(P_d/P_s)^{1/n} \quad \ldots\ldots\ldots\ldots [1]$$

This cylinder volumetric efficiency, as evident from the equation, decreases with the increase in pressure ratio and is reduced still further by the practical effects of valve drops and suction gas heating.

Fig. 7, based on a fixed suction pressure, shows the general variation of the hourly mass of gas pumped versus gas delivery pressure, for three volumetric-efficiency conditions: (*a*) E_v = 100 per cent (unimpaired displacement rate); (*b*) ideal cylinder volumetric efficiency as per Equation [1]; and (*c*) real volumetric efficiency of a practical machine. Combined with the data of Fig. 6, representing cycle performance, the values of pounds of gas pumped per hour, Fig. 7, result in Fig. 8 showing equivalent hourly heat values representative of practical performance. In this idealized presentation of practical machine performance, it is to be noted that the condenser hourly heat rejection is the sum of the hourly evaporator and compressor work-energy effects.

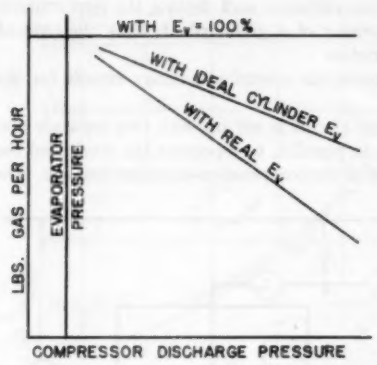

FIG. 7 VOLUMETRIC EFFICIENCY VARIATIONS GOVERNING GAS-PUMPING RATE, VERSUS COMPRESSOR-DISCHARGE PRESSURES

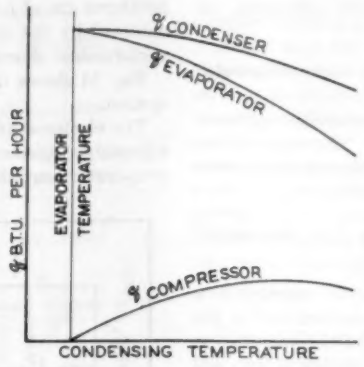

FIG. 8 VARIATION OF HOURLY COMPRESSOR-PLANT ENERGIES WITH DISCHARGE CONDITIONS

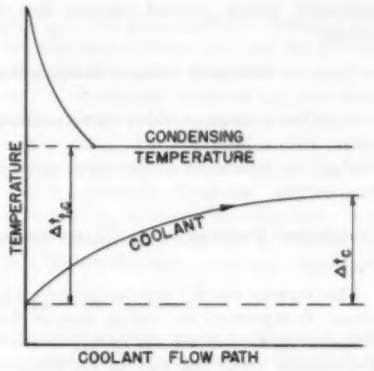

FIG. 9 CONDENSER-TEMPERATURE RELATIONSHIPS VERSUS COOLANT-FLOW PATH

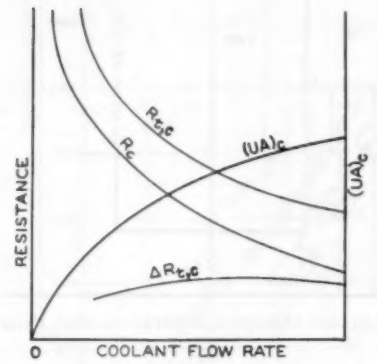

FIG. 10 CONDENSER-THERMAL RESISTANCES VERSUS COOLANT-FLOW RATE W_c

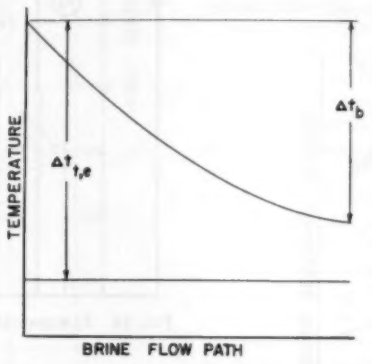

FIG. 11 EVAPORATOR-TEMPERATURE RELATIONSHIPS VERSUS BRINE-FLOW PATH

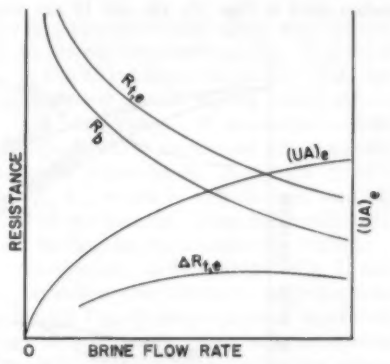

FIG. 12 EVAPORATOR-THERMAL RESISTANCES VERSUS BRINE-FLOW RATE W_b

RESISTANCE CONCEPT APPLIED TO THERMAL-ENERGY CIRCUITS

Condenser Heat Transfer. The flow of heat in a refrigerating-plant condenser may be considered to vary with the over-all temperature difference between the saturation condensing temperature and the inlet coolant temperature, with the effect of the superheat on the condenser performance disregarded. The pertinent temperature relationships are shown in Fig. 9. The hourly heat transfer, as given in a previous paper (5), may be represented in terms of resistance

$$q = \Delta t_{t.c}/R_{t.c} \quad [2]$$

$$R_{t.c} = R_c/(1 - 1/e^{R_c/RU_{A,c}}) \quad [3]$$

$$R_c = 1/(W_c c_c) \quad [4]$$

$$\Delta R_{t.c} = R_{t.c} - R_c \quad [5]$$

These relationships are shown on a general basis in Fig. 10 as plotted against coolant-flow rate W_c.

Evaporator Heat Transfer. The temperature relationships for the evaporator are shown in Fig. 11. As indicated in the earlier paper (5), and similarly as for the condenser, the hourly heat transfer may be shown in terms of resistance

$$q = \Delta t_{t.e}/R_{t.e} \quad [6]$$

$$R_{t.e} = R_b(1 - 1/e^{R_b/RU_{A,e}}) \quad [7]$$

$$R_b = 1/W_b c_b \quad [8]$$

$$\Delta R_{t.e} = R_{t.e} - R_b \quad [9]$$

The general relationships for the evaporator are shown in Fig. 12 plotted against brine-flow rate W_b.

Machine-Energy Transport. In further development of the energy relationships for the compressor plant as illustrated in Fig. 8, and similarly as the resistance concept has been extended to cover heat-exchanger energy flow via the over-all temperature difference (5), so has it been extended further to encompass energy flow in a compressor (7). Equivalent resistance values are used to represent the compressor-plant performance in the present system.

With the variation of compressor performance expressed in terms of over-all temperature difference, as illustrated in Fig. 8 for a fixed evaporator pressure and its corresponding temperature, the resistance values for the compressor energy may be obtained by dividing the temperature difference by the hourly heat quantities

$$R = \Delta t/q \quad [10]$$

Fig. 13 shows the compressor energies with the corresponding

resistance values plotted against temperature difference, as follows:

R_{evap} = resistance value corresponding to hourly evaporator heat

R_{cond} = resistance value corresponding to hourly condenser heat

R_{work} = resistance value corresponding to hourly compressor work

Composite Performance Characteristics Via Electrical Analogy

The various energy resistances may be coupled together in a circuit to represent the energy flow in the condenser and in the evaporator, as well as the energy transport in the compressor. Translation from thermal to electric-circuit values is then advantageous for practical manipulations; the electrical-analogy circuit may be constructed to facilitate evaluation of refrigeration-system performance under varied operating conditions. Herein electrical-resistance values proportional to the thermal resistance values cited in Figs. 10, 12, and 13 are employed, with electric

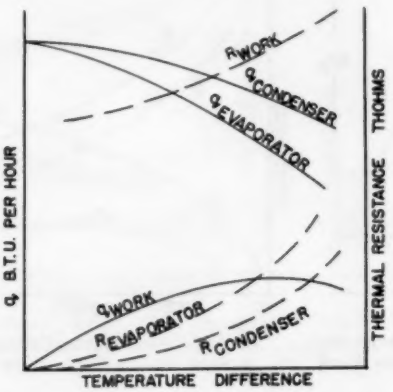

Fig. 13 Compressor Energies and Corresponding Resistance Values Versus Temperature Difference

voltage as the counterpart of temperature difference. (It may be noted that a variable-voltage electric generator might be used to represent the compression-machine characteristics of varying effective temperature difference; the simplest approach, however, seems to be the use of an impressed over-all voltage, as on the condenser system.)

It must be recognized that whatever energy is discharged from the compression machine as transport energy must in turn flow through the condenser system. This machine energy, to reiterate, is made up of the heat flow of the evaporator augmented by the work energy of compression. The extent of the temperature increase above coolant-inlet temperature, as required by resultant condenser heat flow, will in turn set the condensing-vapor temperature, thus, in effect, the compressor delivery pressure.

Similarly, in terms of the brine conditions of temperature and flow rate, the requisite temperature difference necessary for the evaporator heat transfer must thus establish the evaporator temperature and hence its corresponding pressure, i.e., the suction pressure of the compressor plant. The actual compressor energy, as dependent on real volumetric efficiency and hourly displacement, Fig. 8, is dependent on the terminal pressures of the system, thus on the condensing pressure (as determined by the condenser heat transfer) and on the evaporator pressure (as determined by the evaporator heat transfer).

Thus it is apparent that the coolant and the brine initial temperatures are of prime importance and dictate the performance achieved by the compressor of a given plant, thus the over-all performance characteristics.

Fig. 14 shows the complete electrical-analogy circuit for the system.

The electrical-analogy circuit is set up with two separate and adjustable resistances, in parallel, to represent the work and the evaporator heat phases of the compression-machine system. The

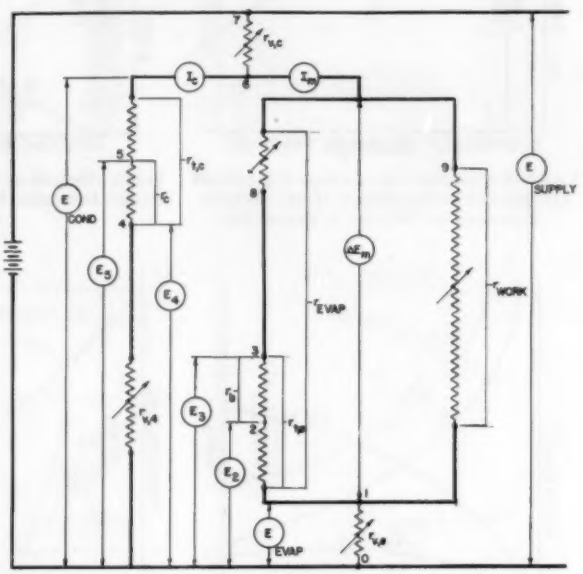

Fig. 14 Electrical-Analogy Circuit for Refrigerating Plant

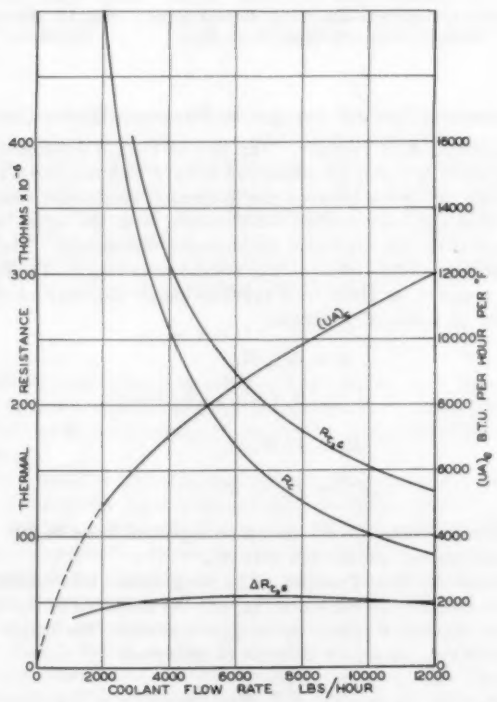

Fig. 15 Illustrative Example of Condenser Characteristics Versus Coolant-Flow Rate W_c

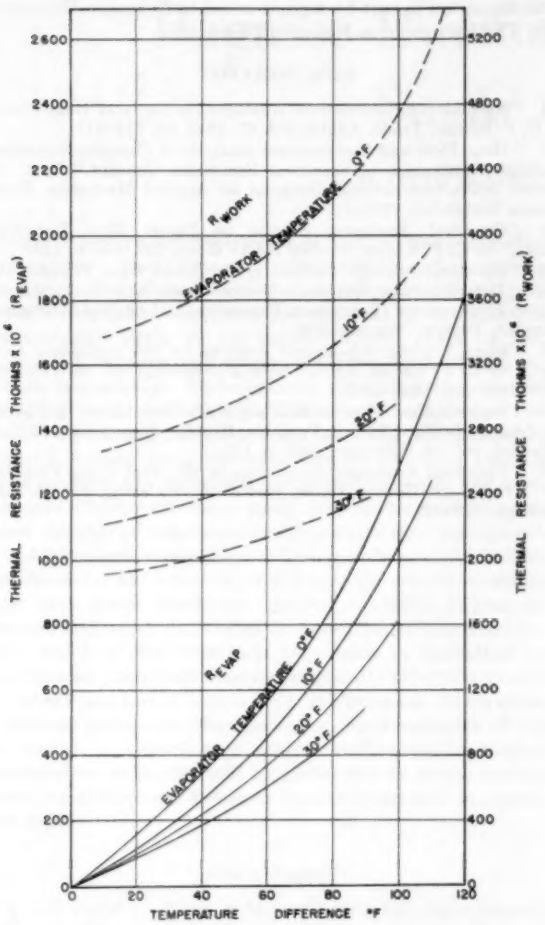

FIG. 16 ILLUSTRATIVE EXAMPLE OF COMPRESSION-MACHINE CHARACTERISTICS VERSUS EQUIVALENT TEMPERATURE DIFFERENCES FOR DIFFERENT EVAPORATOR TEMPERATURES

total current contributed by these two branches must equal that flowing through the condenser circuit, with the top potential common to both condenser and machine sections. The potential value corresponding to the coolant-inlet temperature at point 4 is maintained on the condenser circuit by means of an adjustable ballast resistance $r_{v.4}$. The evaporator heat-transfer section, the counterpart of the condenser heat-transfer section, may be incorporated into the evaporator-heat branch of the machine section.

Both the evaporator and the condenser heat-transfer total resistances $r_{t.e}$ and $r_{t.c}$ include their respective fluid-exchange resistances r_b and r_s. Particularly to be noted is that the brine-inlet temperature equivalent potential at point 3 is to be maintained at some specified value, similarly as in the case of the condenser coolant-inlet temperature. Control for this is in terms of an adjustable "evaporator" ballast resistance $r_{v.e}$ governing the bottom evaporator potential and set in accordance with the requisite brine-inlet potential. The resultant potential difference ΔE_m between the evaporator and condenser-vapor potentials, representative of the compression-machine pressure span, is in turn utilized to determine the effective resistances of the compressor elements r_{evap} and r_{work}.

Adjustment of the analogy circuit is made progressively by changing the condensing-vapor potential through the variable resistance $r_{v.c}$ to keep the condenser current I_c equal to the compression-machine current I_m, while simultaneously keeping the potential at point 4 (coolant-inlet temperature) at its proper value, along with the potential at point 3 (brine-inlet temperature) at its specified value. Coincidental values of r_{evap} and r_{work} are to be maintained at their proper values, as dictated by the machine-potential difference ΔE_m. The current meters show the equivalent energy flowing at given points, and potentials, the equivalent temperatures of the different elements. Adjustment of this type of heat-power analog, it may be recognized, is a matter of manual dexterity, somewhat simplified by the use of automatic equipment, such as, for example, a voltage regulator.

ILLUSTRATIVE EXAMPLE

Figs. 15 and 16 show data for a given vapor-compression system: Compressor displacement, 37.3 cfm; condenser heat-transfer surface, 56.2 sq ft; evaporator heat-transfer surface, 49.1 sq ft. Condenser coolant-inlet temperature is fixed at 70 F and its flow rate W_c at 6000 lb per hr. Plant-performance predictions in terms of varying brine-inlet temperatures are desired for a constant brine-flow rate W_b.

Resistance values for the condenser to correspond with those of Equations [3], [4], [5] are directly obtained from Fig. 15, typified originally by Fig. 12. In Fig. 15 values for the product $(UA)_c$ are included, all values being plotted against W_s, the hourly rate of condenser water flow. The brine-flow rate W_b is constant at 8000 lb per hr, c_b (specific heat) at 0.70 to correspond with a specific gravity of 1.2. In terms of Equation [8], $R_b = 179 \times 10^{-6}$ thohms; via Equation [7], $R_{t.s} = 226 \times 10^{-6}$ thohms; and via Equation [9], $\Delta R_{t.s} = 47 \times 10^{-6}$ thohms. These brine values are the counterpart of those depicted for the condenser in Fig. 15.

Taking real volumetric efficiency, as illustrated in Fig. 7, into consideration, hourly heat values with different evaporator temperatures are used to develop Fig. 16 plotted against equivalent temperature differences between evaporating and condensing conditions, as typified by Fig. 13. Equivalent resistance values, R_{evap} and R_{work}, for the compressor-machine performance in ac-

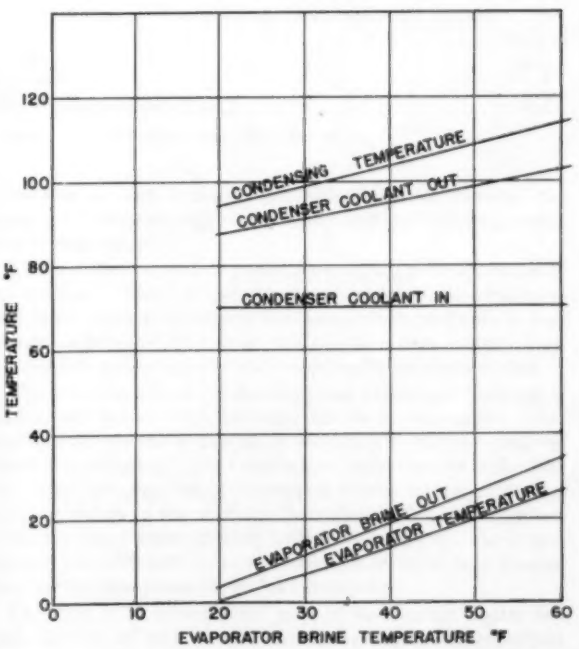

FIG. 17 ILLUSTRATIVE EXAMPLE OF REFRIGERATING-PLANT PERFORMANCE PREDICTIONS VERSUS BRINE-INLET TEMPERATURE

cordance with Equation [10], are specifically shown in Fig. 16 as thohms.

Electric-circuit values, as translated thus from Figs. 15 and 16 and the foregoing evaporator-brine data, for use in the analog arrangement depicted in Fig. 14, are on the basis of 1 ohm electrical resistance = 10^{-6} thermal-resistance thohms, 1 volt = 10 deg F, and 1 milliampere = 1000 Btu/hr.

Fig. 17 shows the variation of condenser-coolant temperatures, in and out (as derived from potentials at 4 and 5), condensing temperature (E_{cond}), brine temperature out (E_2), and evaporator temperature (E_{evap}), all plotted against brine-inlet temperature as established at point 3 (E_3) and varied in steps to determine the plant operating characteristics. The bottom electrode potential is assumed to represent 0 deg F.

Acknowledgment

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BIBLIOGRAPHY

1 "An Electrical Geometrical Analogue for Complex Heat Flow," by C. F. Kayan, Trans. ASME, vol. 67, 1945, pp. 713–718.

2 "Heat Flow and Temperature Analysis of Complex Structures Through Application of Electrical Resistance Concept," by C. F. Kayan, Sixth International Congress for Applied Mechanics, Paris, France, September, 1946.

3 "Electrical Analogger Studies on Panels With Imbedded Tubes," by C. F. Kayan, *Journal ASHVE*, vol. 22, 1950, p. 123.

4 "Electrical Analogy Analysis of Combined Heat Transfer and Vapor Diffusion—the Wet Bulb Thermometer," by C. F. Kayan, Eighth International Congress on Theoretical and Applied Mechanics, Istanbul, Turkey, August, 1952.

5 "Heat Transfer Equipment Analysis by the Resistance Concept," by C. F. Kayan, *Refrigerating Engineering*, vol. 59, 1951, pp. 1195–1199 and 1240–1241.

6 "Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method," by V. Paschkis, *Heating, Piping, and Air Conditioning*, vol. 14, February, 1942, p. 133.

7 "Electrical Analogger Application to the Heat Pump Process," by C. F. Kayan, *Heating, Piping, and Air Conditioning*, vol. 25, July, 1953, pp. 123–127.

Bagasse-Burning in the Mexican Sugar Industry

By OTTO DE LORENZI,[1] NEW YORK, N. Y.

Bagasse is the principal source of heat for the production of raw sugar. Early furnace designs were satisfactory from the standpoint of operation and steam output during the period when labor was plentiful and this by-product fuel was assigned little or no value. Rising costs of fuel, equipment, repairs, and wages, however, have altered this picture materially. Older plants consisted in a multiplicity of small boilers with refractory hearth-type furnaces. Operating efficiency was low, and steam output dropped during long periodic hearth-cleaning periods. These designs are definitely outmoded through the rapid advances which have been made during the past 10 years. Newer designs of steam-generating units now incorporate spreader stokers, water-cooled furnaces, bent-tube boilers, superheaters, air heaters, and bagacillo-return systems. They have more flexibility, greater capacity, higher efficiency, and are less subject to operational interruption. Many units of this type have now been in operation for considerable periods of time in practically all of the world's raw-sugar-producing areas. It is the purpose of this paper to discuss, primarily, the operating characteristics of this new design as experienced with installations in Mexico. Comparative data also will be presented to stress normal operating differences between hearth-type and spreader-fired unit.

Introduction

RAW sugar produced in Mexico during 1951 approximated 835,000 tons.[2] By comparison, this is a relatively small percentage of world-wide production. The important fact, however, is that this tonnage is more than adequate to satisfy the relatively high domestic sugar-consumption rate of 63.4 lb per capita. Assuming an average yield of approximately 16 tons of sugar per acre, there are then over 52,000 acres of fertile land in Mexico devoted to growing cane.

The dry substances of cane are essentially 56 per cent sugars and 44 per cent fiber. Thus the quantity of dry bagasse produced in Mexico, based on 1951 sugar yield, is over 650,000 tons per year. The average heating value of dry bagasse is approximately 8200 Btu per lb. Under normal operating conditions an efficient mill needs only 85 to 90 per cent of the bagasse it produces for the extraction of raw sugar. Mills which additionally operate white-sugar refineries and/or distilleries will require considerable quantities of purchased fuel. In some areas increasing amounts of auxiliary fuel are being used for the production of raw sugar. The bagasse thus replaced may be used for the manufacture of pulp, purified cellulose, fine paper, newsprint, wallboard, and other similar processes in which formerly the basic raw material was wood.

These expanding uses for bagasse together with a diminishing labor supply, as well as rising cost of auxiliary fuel, equipment, repairs, and wages, have made it necessary to modernize or replace older steam-generating installations and also to reduce heat losses in other mill areas. Modernization programs, involving applications of newer type of fuel-burning equipment to existing boilers and/or the installation of complete modern steam-generating units, have been undertaken by a number of Mexican sugar mills. Some of these units have now been in service for over 3 years, and thus performance characteristics may be compared with those of earlier design. Before making such comparisons, however, it is necessary to consider the fundamentals involved in burning a wet cellulose-type fuel such as bagasse.

Combustion of Bagasse

Bagasse is a fuel of variable moisture content which, in turn, is a direct function of mill operation. It may vary from 30 per cent to well over 60 per cent. For average conditions it will be approximately 45 per cent. The proximate analysis depends upon the degree to which sugar extraction, from cane, is carried as well as on the locality in which the cane is grown. Data for a typical Mexican bagasse from cane grown in the vicinity of Vera Cruz are given in Table 1.

TABLE 1 PROXIMATE ANALYSIS (DRY BASIS)

	Per cent
Volatile matter	87.0
Fixed carbon	11.6
Ash	1.4
Sulphur (separately determined)	0.6

NOTE: Higher heating value, 8143 Btu per lb.

Because of high moisture and volatile-matter content, the process of combustion consists of three consecutive and somewhat overlapping stages.

The first stage is that of preliminary drying or evaporation of the moisture. This is a heat-absorbing process in which evaporation takes place at relatively low temperature and only as fast as heat is supplied to change the moisture into steam. Fuel temperature will not exceed 212 F until moisture is evaporated.

The second stage is one of distillation and burning of the volatile matter. It occurs after moisture has been evaporated fully, and addition of heat to the fuel is continued so that its temperature will be raised to a point where the volatile matter is distilled off. Air is then supplied to the gaseous volatile matter, and when the temperature of this mixture, through the continued addition of heat, reaches approximately 1100 F, it bursts into flame and burns. Thus the first phase of distillation is one of heat absorption and the final phase one of heat generation.

The third stage occurs when most of the volatile matter has been distilled off and the surface of the residual fixed carbon reaches a glowing temperature and finally burns as air is brought into contact with it.

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[1] Director of Education and Fuels Consultant, Combustion Engineering, Inc. Fellow ASME.

[2] "Sugar Reference Book," Vol. XXI, 1953.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters on November 18, 1953. Paper No. 54—Mex-4.

Two-Stage Furnace Designs

Inasmuch as volatile matter constitutes over 85 per cent of the combustible products in dry bagasse, practically all burning will take place away from the fuel bed. It is for this reason that earlier furnace designs were of the two-stage type. The primary furnace consisted of a Dutch oven equipped with either sloping or stepped grates, flat grate, or hearth of the horseshoe or Cook type. Fuel was introduced continuously through roof openings onto the grate or hearth, where it formed into conical piles. Drying, gasification, and some burning of residual carbon take place on the surface of the pile. Because of this, practically all of the air for combustion is supplied over the fuel bed instead of through it. Since this method of combustion depends, in its early stages, on heat absorption, the use of refractory arches and walls in the primary furnace is of considerable importance. Their function is to provide maximum possible amount of radiant and reflected heat for drying, gasification, and preventing the fire from being extinguished, even though the principle of showering fresh wet bagasse continuously onto the surface of the cone-shaped pile is used. The gaseous products of distillation sweep over the surface of the fuel pile and are mixed with air so that combustion is completed after they enter the secondary furnace.

Automatic control systems are not well adapted to two-stage furnace operation, and therefore manual regulation of fuel and air supply is employed. Excess air varies over a wide range because of difficulty in proportioning it closely to fuel-burning rate. Consequently, over-all efficiency is low. Furthermore, there is the necessity in many installations for taking the boiler out of service periodically to remove clinkers and slag from grates or hearth, and fly ash from the boiler passes. These outages, usually at weekly intervals, result in low average boiler output. Additional units must, therefore, be kept in stand-by service to insure continuity of production.

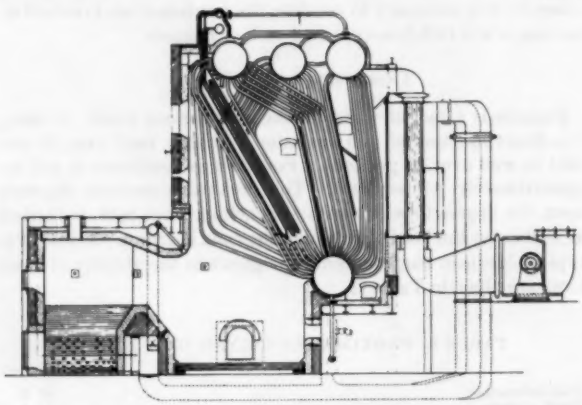

Fig. 1 C-E VA Boiler With Two-Stage Furnace Designed for Bagasse-Firing
(Primary furnace composed of three refractory horseshoe hearths; secondary furnace sidewalls are partially water-cooled.)

During 1947 the unit shown in Fig. 1 was placed in service for a large Cuban sugar mill. The furnace employed is of the two-stage Cook type. Primary furnace, comprising three horseshoe hearths, is of all-refractory construction. The large secondary furnace is also of refractory construction except for a portion of each side wall which is water-cooled. Three oil burners are installed in the rear wall of the secondary furnace. A tubular-type preheater supplies air for combustion at 320 F. Maximum guaranteed capacity, when burning Cuban bagasse containing 50 per cent moisture, is 72,500 lb of steam per hr at 160 psi and 520 F at a predicted efficiency of 66.4 per cent.

Performance of this unit has been very satisfactory. It is, however, typical of that associated with the better design of two-stage furnaces employing pile burning. Charts of steam flow and CO_2 over a normal 24-hr day are shown in Fig. 2. From these it will be noted that capacity is maintained relatively constant, from 75,000 to 85,000 lb per hr, for two periods of approximately 5 hr each. Corresponding CO_2 readings are at a high level (practically off the chart owing to incorrect zero setting).

For the remaining 14 hr, capacity and CO_2 fluctuate widely. Some of this fluctuation may be attributed to changes in rate of fuel feed, although an ample reserve is available from the pile in the furnace. The principal cause, however, results from a gradual accumulation of refuse on the hearths. These must be cleaned periodically to maintain a reasonably high 24-hr steam output.

Total elapsed time for refuse removal, as indicated on the chart, is 3 hr 40 min, divided into three periods: A (4:40 a.m. to 5:35 a.m.); B (11:40 a.m. to 1:20 p.m.); and C (7:45 p.m. to 8:50 p.m.). Time required to clean each one of the three hearths at period A is clearly shown on both the steam-flow and CO_2 charts. Effect of load or fuel-feed fluctuations on CO_2 with two-stage furnaces is shown over the period 7:30 a.m. to 10:15 a.m.

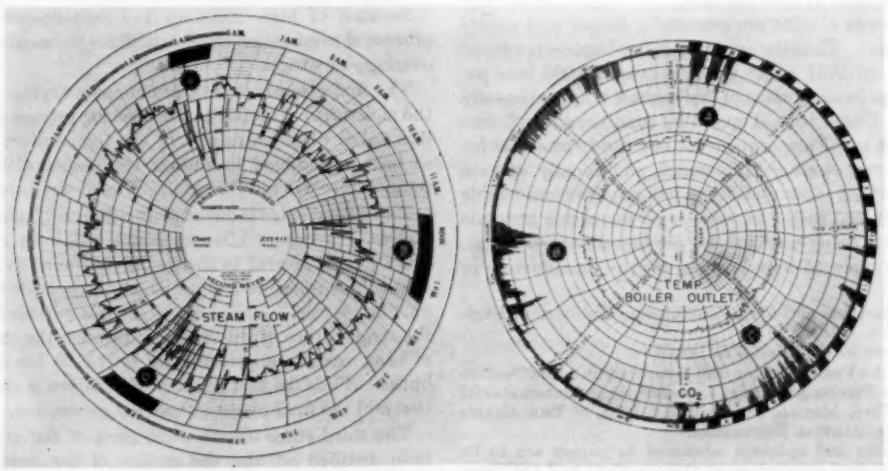

Fig. 2 Steam-Flow and CO_2 Charts for Bagasse-Firing of Two-Stage Furnace Design Shown in Fig. 1

The most recent developments in two-stage furnaces are the designs of Ward and Martin. In these the primary furnace is located directly below the secondary furnace. Fuel is supplied through sloping spouts in the front wall and falls onto conical piles on the horseshoe-shaped hearths. Air is admitted overfire around the pile through wall tuyères. Arches are employed to form a restricted throat opening through which the mixture of air and gasified combustible products flows at high velocity into the secondary furnace. With this arrangement there is considerable carry-over of partially burned bagasse fines which frequently results in severe slagging of boiler tubes and char discharge out of the stacks. Hearth conditions are no different from earlier Dutch-oven types, and thus outages for periodic removal of slag are still necessary.

Single-Stage Furnace Designs

About 1908, Joseph W. Hays was credited with coining a phrase which has been time-proved throughout the entire period of modern stoker development. He said: "Fire thin and often." In the case of bagasse-furnace operation this was apparently overlooked until about 1946. At that time an analysis indicated the principles of spreader-stoker firing were well-suited for burning bagasse in a single-stage furnace. With an installation of this type, it was reasoned that flash drying of moisture and distillation of a large portion of volatile matter would occur during flight of fuel particles, through the furnace atmosphere, from distributor to grate. Any remaining volatile matter together with fixed carbon would then be burned in the thin fuel bed at grate level.

A trial installation, under an HRT boiler, quickly confirmed this analysis as a practical one and simultaneously indicated the necessity for providing a more suitable means to regulate bagasse flow to the spreader unit. A variable-speed rotary feeder, consisting of a hopper and metering drums, was then developed to replace original designs. With this arrangement it becomes possible to control closely the rate of bagasse feed and thereby to either maintain constant steam output or quickly follow load variations as they occur.

During 1947 a complete steam-generating unit incorporating these ideas was installed in the same Cuban sugar mill in which is also the unit shown in Fig. 1, and placed in service for the 1947-1948 grinding season. This unit, designed for a continuous capacity of 60,000 lb per hr at 160 psi and 520 F at an efficiency of 68.3 per cent, is shown in Fig. 3. It was kept in continuous operation, except for the normal Easter holiday and mill shutdown caused by rain, until end of the crop.

The single-stage furnace eliminated Dutch-oven extensions, partition walls, and other special requirements of two-stage designs. Side-wall and waterwall cooling were used to improve furnace availability and thereby make it possible to keep the unit in service continuously throughout the grinding season.

Two drum-type feeder assemblies are located directly below the bagasse conveyer. The rotary drums discharge metered bagasse through inclined chutes to four mechanical-type spreader units. These units, in turn, project the fuel into the furnace in a thin, uniform, and widely dispersed stream, where it is dried and partly burned in suspension and the remainder consumed after dropping to the grate. The fuel bed is thin, uniform, and actively burning over the entire grate, and there are no piles or dead areas. The dumping-type grate surface is 9 ft long × 18 ft wide and extends across the length and width of the furnace, without intervening partition walls. Preheated air is supplied

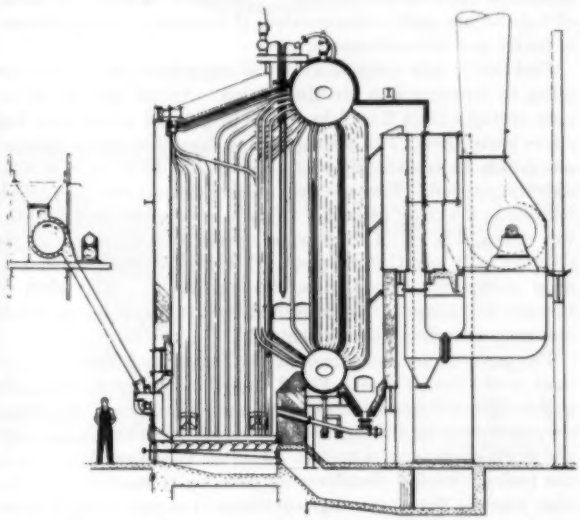

Fig. 3 C-E Steam-Generating Unit With Single-Stage Water-Cooled Furnace for Spreader-Stoker-Firing of Bagasse

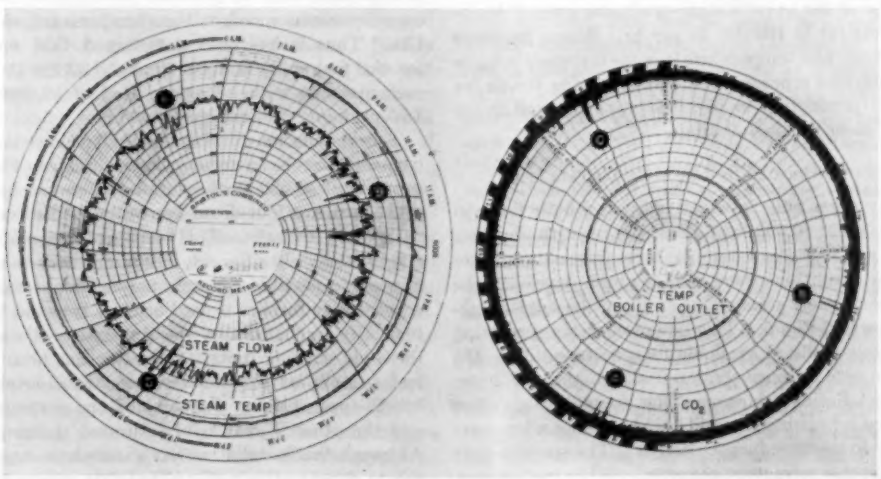

Fig. 4 Steam-Flow and CO_2 Charts for Bagasse-Firing of Single-Stage Furnace Design Shown in Fig. 3

to the fuel bed from an undergrate plenum chamber which also serves as an ashpit. The grate, as well as plenum chamber, is sectionalized laterally, corresponding with the number of spreader units. Each air zone is under individual damper control, and therefore fires can be cleaned one section at a time without changing rate of bagasse feed. Power cylinders, operated by either compressed air or steam, are used to lower and raise the grates during the dumping operation. A tubular-type air preheater raises the temperature of air for combustion to over 275 F. Preheated air is not essential to successful operation, but it is a desirable means for increasing over-all efficiency and promoting more rapid combustion of the fuel.

How well performance has measured up with guarantee and predictions is best illustrated by quoting some actual operating experiences. Continuous capacity is usually maintained between 75,000 to 80,000 lb per hr with peaks of 85,000 lb per hr, and these are limited only by induced-draft-fan capacity. CO_2 is maintained between 16 and 17 per cent, which corresponds with 20 to 15 per cent excess air. Operating efficiency continually averages between 67 and 68 per cent. Variations in moisture content of bagasse from 45 to 55 per cent have no noticeable effect in the furnace. Changes in quality and sizing of bagasse are easily compensated, if necessary, by adjustment of feeder and spreader speeds.

Fuel bed is thin and uniform at all capacities with little or no piling to interfere with air distribution. Actual fuel-bed thickness averages from 2 to 3 in. It is bright and active over the entire grate area. The ash supporting the active fuel bed gradually builds up to a thickness of approximately 2 in. over an 8-hr operating period. Fires are then quickly cleaned with little or no noticeable effect on capacity. This is clearly demonstrated on the charts, Fig. 4. Grates were cleaned at A (4:30 a.m.); B (10:45 a.m.); and C (8:00 p.m.). There was only a momentary drop in capacity during these cleaning periods. The effect of furnace conditions is best measured by the CO_2 readings which also were disturbed only momentarily.

For performance comparison with the two-stage furnace, refer back to the similar charts, Fig. 2, from another unit in this same plant. The differences in capacity and CO_2 are only two of the several reasons for the higher sustained efficiency obtainable with the single-stage furnace design. Another important difference is the periodic weekly shutdown for cleaning of the two-stage design, whereas the single-stage continues in service without interruption until there is a mill shutdown.

The design shown in Fig. 3 has been developed primarily for capacities above 40,000 lb steam per hr. A large number are now operating in many of the world's sugar-producing areas. Maximum capacity thus far is 100,000 lb per hr. Steam pressures have reached 650 psi, and temperature in some instances is up to 750 F. The conditions represent the over-all design trends for steam and power production in new operations as well as for modernization of older mill installations.

Performance of Mexican Sugar-Mill Installations

There are many sugar mills in which individual units range in capacity from 20,000 to 50,000 lb of steam per hr, with pressures from 125 to 180 psi, and steam temperatures from saturated to 500 F. For these conditions there also has been developed the economical and completely standardized unit, of the single-stage-furnace type, shown in Fig. 5. This design of steam-generating unit, including waterwalls, is available for pressures up to 475 psi and has been used extensively for coal, oil, and gas firing. With bagasse installations, however, the normal combustion volume is enlarged by raising the unit and providing a lower refractory chamber to increase furnace height. The spreader unit is then installed in this refractory chamber. Oil or gas burners, where required, are placed in their usual front-wall position above side-wall header level. Units of this type have been installed at

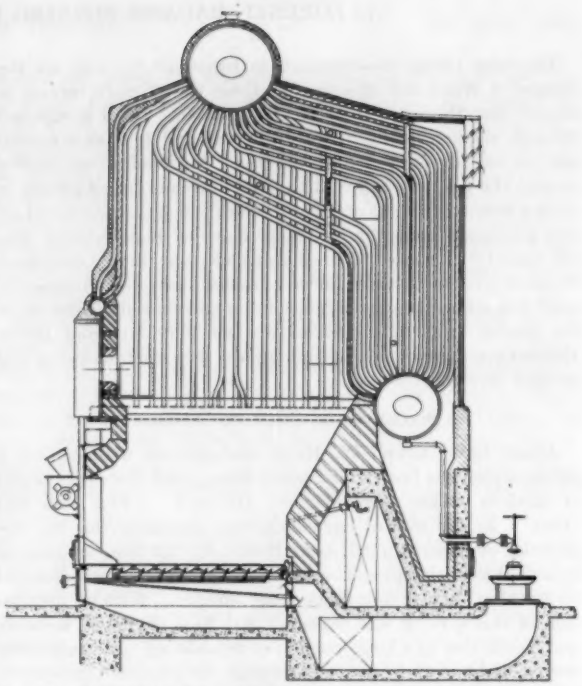

Fig. 5 C-E VU-10 Steam-Generating Unit for Single-Stage Bagasse-Stoker-Firing in Lower Refractory Furnace and/or Oil-Firing in Upper Water-Cooled Furnace

Ingenio de Casasano, S. A., in the vicinity of Mexico City, and at Ingenio Independencia near Vera Cruz. For each of these, capacity is 30,000 lb of saturated steam per hr at 180 psi.

The boiler plant at Casasano consists of one steam-generating unit, Fig. 5, equipped with bagasse stoker, oil burners, and tubular air preheater. A second unit, similar to the first one, is equipped with oil burners only; its setting height, however, is sufficient for possible future installation of a bagasse stoker. The third unit consists of a longitudinal three-drum sectional-header boiler with three horseshoe furnaces in a Dutch-oven setting, and a secondary combustion chamber located immediately below the boiler bank. Oil may be fired by inserting nozzles through the rear wall of the secondary combustion chamber.

Owing to absence of flowmeters and CO_2 recorders at Casasano, it is possible only to estimate performance from known feeder capacity versus speed, exit-gas temperature, and snap samples of CO_2. Thus it has been determined that operating capacity, for the first unit, is approximately 32,000 lb of steam per hr, corresponding with a bagasse feed of 12,600 lb per hr; CO_2, 15.5 per cent; and exit gas at 460 F.

The installation for Ingenio Independencia consists of three bagasse-stoker-fired units. One is similar to Fig. 5, with a design capacity of 30,000 lb of saturated steam per hr at 180 psi. The other two are four-drum bent-tube boilers originally provided with Ward furnaces. This two-stage design is shown in Fig. 6A, and immediately adjacent, in Fig. 6B, may be seen the simplification made possible through use of single-stage spreader-firing. A refractory furnace is used because it would have been difficult to justify, economically, the application of waterwall cooling to these old existing units. Operation has been more satisfactory and reliable, at a better level of steam output, than with the hearth-type design. Capacity of one revamped unit is 33,000 and the other 27,500 lb of saturated steam per hr at 150 psi. Although this installation has a complete complement of instruments, comparative performance data were not readily available.

Ingenio El Potrero, S. A., also in the vicinity of Vera Cruz, is another interesting and relatively large installation. At present

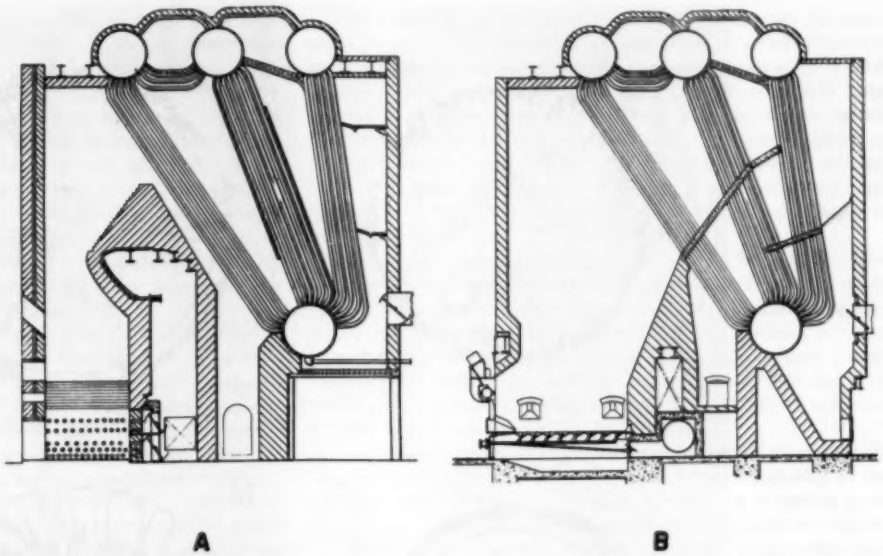

FIG. 6A STIRLING BOILER ORIGINALLY EQUIPPED WITH WARD TWO-STAGE FURNACES

FIG. 6B SAME STIRLING BOILER WITH MODERNIZED SINGLE-STAGE FURNACE AND BAGASSE-STOKER FIRING

A B

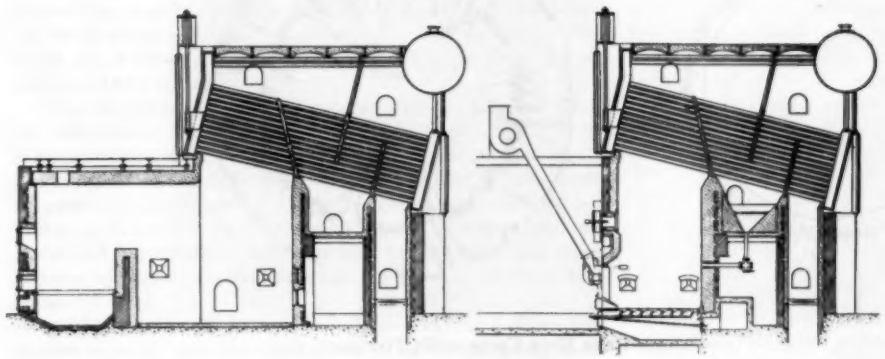

FIG. 7A C-E BOX-HEADER BOILER AS ORIGINALLY EQUIPPED WITH FLAT GRATES AND TWO-STAGE FURNACE

FIG. 7B SAME C-E BOILER WITH MODERNIZED SINGLE-STAGE FURNACE AND BAGASSE-STOKER-FIRING

A B

there are eight boilers in service, seven of which are bagasse-fired and the eighth oil-fired.

Boilers Nos. 1 and 2 are box-header types, set in battery and now provided with Martin two-stage hearth furnaces. Boilers Nos. 3, 4, 5, and 6 are also box-header types, set in two batteries and equipped with single-stage furnaces and bagasse stokers. All six units originally were installed with flat grates set in Dutch ovens, as shown in Fig. 7A. The present application of bagasse stokers to four of these units is shown in Fig. 7B.

Boiler No. 7 is a cross-drum sectional-header type having two Ward furnaces and also an air heater. Recently this unit also was equipped with two drum-type bagasse feeders. With this arrangement it is therefore possible to regulate flow of fuel to the hearth and thereby maintain more uniform steam generation between cleaning periods.

Boiler No. 8 is of the design shown in Fig. 5, but low-set for oil-firing only.

El Potrero is one of the few sugar mills in Mexico equipped with sufficient instruments to provide reliable operating records. A series of four interesting charts, Fig. 8, indicate the comparative steam-generating characteristics of the four different types of fuel-burning application. Normally, all the bagasse-fired units are adjusted for constant capacity so that any changes in steam demand are handled by the oil-fired unit.

Chart I, Fig. 8, for boiler No. 2 with Martin furnace, shows the fluctuation which may be expected with pile burning in a two-stage furnace. Fuel flow to hearth is unregulated, and consequently steam generation varies accordingly. Drop in capacity due to cleaning of hearths occurs at 2:15 p.m. to 3:15 p.m.; 10:20 p.m. to 11:45 p.m.; and 7:45 a.m. to 8:30 a.m. It will be noted that two dips in steam output are usually recorded for each cleaning period, one for each of two hearths per unit.

Chart II, Fig. 8, is from boiler No. 4, which is equipped with a single-stage furnace and bagasse stoker. Steam output is practically constant around the clock with frequent minor variations owing to change in rate of feedwater flow to the boiler. The three cleaning periods, 3:40 p.m. to 3:50 p.m.; 11:05 p.m. to 11:10 p.m.; and 5:55 a.m. to 6:05 a.m., have only slight and momentary effect on capacity.

Chart III, Fig. 8, is from boiler No. 7 and represents characteristic performance for two-stage Ward-furnace operation. In this case, however, fluctuations are reduced somewhat in comparison with those resulting from usual pile-burning, because a drum-type metering feeder is used to regulate fuel flow. Burning rate, however, still varies widely because it is almost impossible to control air flow closely over the fuel-pile surface. Cleaning periods at 1:55 p.m. to 2:25 p.m.; 9:50 p.m. to 10:15 p.m.; and 5:55 a.m. to 6:45 a.m. are indicated by deep drops in capac-

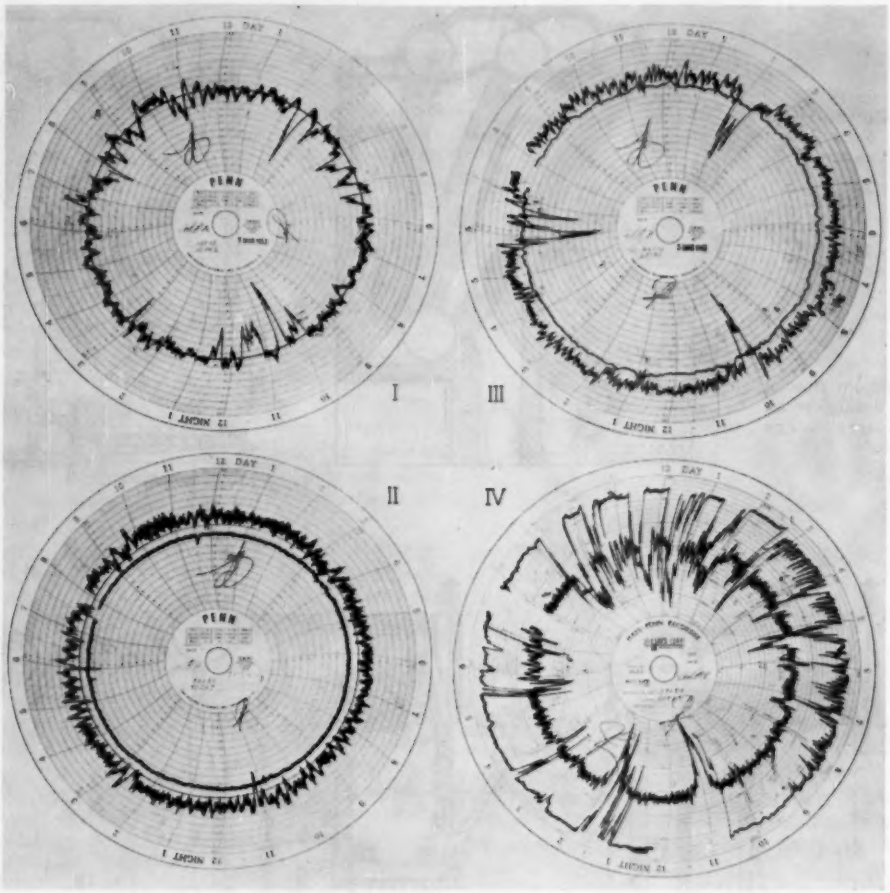

FIG. 8 STEAM-FLOW CHARTS FROM FOUR UNITS AT EL POTRERO

I C-E box-header boiler with Martin hearth-type furnaces
II C-E box-header boiler with bagasse stoker
III B&W sectional-header boiler with Ward hearth-type furnaces
IV C-E VU-10 steam-generating unit for oil-firing only

ity. The last one, because of its duration, perhaps indicates heavy slagging on the hearth and may have resulted from insufficient cleaning during the previous period. In this case there is also an indication of the time required to clean each of the two hearths.

Chart IV, Fig. 8, shows the many abrupt changes in capacity occuring during normal mill operation. These are quickly compensated for on boiler No. 8 by regulating the oil-firing rate to maintain uniform pressure on the main steam header.

A number of tests to establish performance characteristics of capacity and efficiency have been conducted on the units at El Potrero. Bagasse is a bulky fuel with low heat content, and to weigh it, for an accurate boiler test, would present difficulties. Inasmuch as the drum-type bagasse feeder, employed with the units, is actually a metering device subject to calibration, it was decided to use it for fuel-weight determination. Accuracy of results obtained, therefore, depends largely upon how close actual feeding rate corresponds with the speed of the bagasse feeder. Actually, rate of feed is not exactly proportional to speed changes. To compensate for this it was decided to maintain the feeder speed constant at one calibration rate for the entire 24-hr test run.

Fortunately, cane-grinding conditions also were relatively constant since no mill chokes occurred, and thus sizing, as well as moisture content of the bagasse, remained reasonably constant. Under these conditions the following averaged data were recorded, and corresponding calculations indicate an efficiency of 59.7 per cent. Inasmuch as sufficient additional data also were collected to measure magnitude of various losses, it is possible to prepare a heat balance and, thereby, provide a more precise check on test accuracy.

Data from the 24-hr test on boiler No. 5 at El Potrero on February 3 and 4, 1950, are given in Table 2.

TABLE 2 EL POTRERO BOILER NO. 5
TEST DATA—24 HR, FEBRUARY 3 AND 4, 1950

Output, steam, lb/hr	30872
Fuel consumed (bagasse), lb/hr	13235
Pressure, steam, psi	151.6
Temperature, feedwater, deg F	181.9
Temperature, exit gas, deg F	630.0
Temperature, air, deg F	78.7
CO_2, per cent	13.5
Fuel Analysis (Proximate):	
Moisture, per cent	49.8
Volatile matter (dry basis), per cent	87.0
Fixed carbon (dry basis), per cent	11.6
Ash (dry basis), per cent	1.4
Sulphur, sep. det. (dry basis), per cent	0.6
Heating value (dry basis), Btu/lb	8143
Heat Balance:	
Efficiency (calculated from metered fuel and steam flow), per cent	59.7
Loss from moisture and H_2 in fuel, per cent	21.7
Loss in dry flue gas, per cent	16.2
Loss from carbon in refuse, per cent	0.2
Loss from radiation, per cent	1.3
Errors and unaccounted-for loss, by difference, per cent	0.9
Total	100.0

Perhaps the item "carbon loss in refuse" may appear somewhat low by comparison with other solid fuel. It should be remembered, however, that percentage of ash in dry bagasse is 1.4 per cent. With spreader-firing, the active portion of the fuel bed is a relatively thin layer of fixed carbon. Grates are cleaned once every 8 hr, and thus the amount of carbon discharge to ashpit is, at most, only an insignificant portion of the actual fuel fired. Fly ash carry-over deposited in boiler hopper is returned to the furnace for reburning, and therefore the only additional loss is the carbon discharged to the atmosphere.

The "radiation-loss" item is obtained from charts prepared by the American Boiler Manufacturers Association based on the construction of furnace and boiler setting used.

The item "errors and unaccounted-for losses" is well within the expected margin for an accurate test. Calibration tests of both bagasse feeder and flowmeter were conducted carefully, and any deviations make up the "errors" item. The unaccounted-for portion represents the carbon loss in fly ash discharged to atmosphere with the stack gas. As previously indicated, this loss should be quite low because ash in dry fuel is only 1.4 per cent.

From the foregoing it therefore may be concluded that an efficiency of 59.7 per cent closely represents the normal performance of boiler No. 5. Other similar tests have been conducted on boilers Nos. 4 and 6. For boiler No. 4 the average capacity was 31,600 lb steam per hr at an efficiency of 59.3 per cent. For boiler No. 6 the capacity was 32,500 lb steam per hr at an efficiency of 61.2 per cent.

The unit also can be operated satisfactorily on fuel oil alone or in combination with bagasse. Full capacity is possible with almost any proportion of the two fuels. On one of these units the bagasse feed and air flow were reduced so that steam output dropped to 18,700 lb per hr. One oil burner was placed in service, and evaporation quickly increased to 30,400 lb per hr. Both oil flame and bagasse fires were clear and bright, indicating good furnace conditions for the simultaneous burning of these two fuels.

Another single-stage-furnace installation, having a somewhat greater capacity than the three discussed in the foregoing, is at Ingenio Tamazula, S. A., Jalisco, Mexico. This steam-generating unit, designed for a capacity of 51,500 lb of steam per hr at 160 psi and 480 F, consists of a four-drum bent-tube boiler similar to Fig. 1, but equipped with a single-stage furnace and bagasse spreader stoker. It has now been in service since 1950, and again, no operating data are available.

Turbulent Suspension Burning

Recently some newer principles have been developed which extended the capacity of steam-generating units in which fibrous cellulose fuels are burned. Designs employing these principles are well illustrated in Fig. 9. They include a towerlike single-stage furnace in which it becomes possible to take full advantage of flash-drying effect by increasing time for flight of fuel through the furnace atmosphere. This is accomplished by changing the location of spreader units to an elevation of from 12 to 20 ft above grate level in place of the usual 4 to 5 ft. Overfire air nozzles are positioned in the four furnace corners below spreader-unit level. With this arrangement fuel projected into the furnace is spun around by high-velocity overfire air streams and retained in the rising column of gas for a greater length of time, before coming to rest at grate level. It is the increased time of flight, through the hottest furnace zone, which causes: moisture to be quickly flashed into vapor; partial or complete devolatilization of the fuel; and residual combustible constituents to become ignited before reaching the grate. Slug-feeding produces no noticeable changes at fuel-bed level, and consequently there are no adverse effects on operating results.

The foregoing principles of "turbulent suspension burning" may, therefore, be defined as an operation in which relatively large particles of fuel are showered downward from some considerable height in a towerlike furnace, through relatively large quantities of highly turbulent gas and overfire air, to effect: Flash drying; partial or complete devolatilization; ignition of fuel constituents; and finally, uniform distribution of residual fuel on the grate surface, in the lower furnace zone, where combustion is completed through the use of reduced percentages of undergrate air.

Large units employing turbulent suspension burning are now in operation with such fuels as wood-room refuse from pulp mills; spent-wood refuse from naval stores plants; sawmill hog fuel comprising constantly varying percentages of sawdust, dry and green planer shaving, hogged wet bark, and pulp chipper refuse; residue from furfurol production; bagasse; and high and low-volatile coal. Capacity range extends from 80,000 to well over 300,000 lb of steam per hr.

This principle will be employed for a unit, almost identical with the one shown in Fig. 9, which is to go into service shortly at the Kekaha Sugar Company's plant in Hawaii. A traveling grate for continuous removal of ash is installed in a furnace having four water-cooled walls. The two-drum single-pass boiler is equipped with an interbank superheater. A tubular preheater is used for increasing temperature of air for combustion. Burners for tangential firing of fuel oil are located at a level somewhat above the spreader unit.

Because Hawaiian sugar cane is harvested mechanically, it produces a bagasse which is difficult to burn effectively. Silt remaining on the root structure mixes with ash in the cane fiber

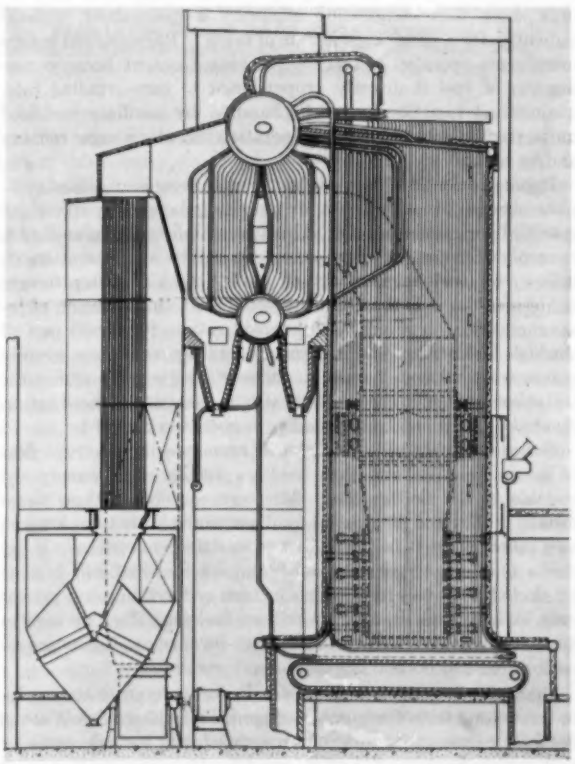

Fig. 9 C-E Steam-Generating Unit Consisting of Water-Cooled Furnace, Single-Pass Boiler, and Tubular Air Heater, Employing Principle of Turbulent Suspension Burning of Hawaiian Bagasse and/or Tangential Firing of Fuel Oil

and produces a fluxing action, at fuel-bed temperature, that results in formation of tenacious and sticky clinkers which, in turn, are the greatest single cause of boiler shutdowns with conventional furnace designs. Combined use of turbulent suspension burning, water-cooled walls, continuous ash discharge, and single-pass boiler will overcome these operating difficulties and provide exceptional reliability, improved use and capacity factors, and long in-service availability.

Conclusion

In concluding the foregoing discussions, it is well to review the reasons contributing to the exceptional success in spreader-firing of bagasse with single-stage furnaces.

Development of the variable-speed drum-type feeder is perhaps the most important contribution to success of bagasse stoker-firing. Unlike other available designs, it does not pack or compress the fibrous fuel to a point where clogging and uncertainty or complete failure of feeding occur.

Of considerable importance is distribution of bagasse over the grate surface in a level and thin layer to insure uniform flow of undergrate air through all fuel-bed sections. This is best accomplished with the mechanical-type spreader unit for a wide range in moisture content, fuel-size consist, and feeding rate. This conclusion results from conducting a series of comparative tests during which many different mechanical and pneumatic spreader designs were employed.

Drum-type metering feeders, mechanical spreader units, and thin active fuel beds provide an ideal combination for the application of automatic control systems. With this arrangement there is rapid response for changes in boiler output to meet variations in plant steam demand. Furthermore, uniform CO_2 records show that combustion efficiency is maintained, without difficulty, for a wide range in rate of firing. Bagasse-fired boilers usually are operated at relatively constant output because production of fuel is directly proportioned to cane-grinding rate. Major load variations must be handled by auxiliary fuel-fired units, particularly where plant operation includes a sugar refinery and/or a distillery.

In practically all of Mexico's sugar mills there is an almost complete absence of instruments for providing day-to-day records of essential performance data. This lack of information, no doubt, is responsible for lowering possible capacity and operation efficiency of most bagasse-burning units. As a first step toward modernization it is therefore suggested that this question of instrumentation be given careful consideration, for it will pay off through fuel savings and improved operating conditions, even in plants with outmoded types of furnace designs. To realize the full effect from such records, it is also necessary that these instruments be maintained, continually, in good working order.

Occasionally a "mill choke" will momentarily interrupt flow of normal bagasse and then send a quantity of extremely wet bagasse to the boiler units. Moisture content at these times usually will exceed 66 per cent, and heating value may be lowered to a point where it is insufficient to sustain combustion. If the choke is of relatively long duration, the fuel bed will become blanketed with wet fuel, which in turn reduces burning rate or even extinguishes the fire. Auxiliary fuel must then be used to meet steam demands until satisfactory furnace conditions are re-established and normal bagasse is again available.

Grates for bagasse stokers are zoned, for undergrate air supply, to correspond with the number of spreader units used. When it becomes necessary to dump accumulated ash, a single spreader unit is usually shut off, and air used for its corresponding grate section is continued until fuel bed is burned down. When burning-down is completed, air flow is also shut off and grates are dumped and immediately returned to operating level. Fuel feed and air supply are then quickly restored to normalize operation. The same procedure is followed for the remaining grate sections until all are cleared. Ashes are not removed from the pits until normal operation is fully re-established for the entire stoker. With this procedure less time and attention are required for necessary periodic cleaning of fires, and consequently there are only minor momentary drops in capacity and CO_2.

Satisfactory performance records for Mexico's bagasse stoker installations are a credit to the progressive managements of these sugar mills. No doubt there are others in which modernization of existing boilers, or purchase of new units, could be undertaken profitably. The savings that would accrue from reduction in quantity of purchased fuel resulting from improved bagasse-burning efficiency, together with reduced outage and maintenance costs coupled with higher sustained in-service and capacity factors, can, no doubt, be capitalized to write-off costs over a reasonably short period. This is a challenge to management, particularly in view of today's increasing cost spiral.

Discussion

H. G. Meissner.[3] As noted in the introduction to the paper, many by-product uses for bagasse are being developed, which raises the question as to the relative merits of this material for fuel or for by-product output. The following items must be considered in such an evaluation:

1 Well over 90 per cent of the heat energy in the steam generated is utilized in evaporating or refining the juice for production of sugar, so that the engines or turbines for mill drives and other power requirements are used as rather efficient reducing valves. Fortunately, the heat balance is such that, in a raw-sugar mill, there is usually a surplus of bagasse, although, when a refinery is included, some auxiliary fuel is required.

2 Bagasse has been used successfully as a raw material for wallboard, paper, furfural, and other by-products. Studies made during the past few years have developed the following cost figures.

Bagasse, as produced in Louisiana, averages 49 per cent moisture, 45 per cent fiber, and 6 per cent soluble solids. About 97 per cent will pass through a 1-in-square-mesh screen. Dry bagasse contains 65 to 70 per cent fiber and the balance pith, which has no fiber strength and may or may not have to be removed, depending on the papermaking or other process involved. Its removal is quite costly.

A 2000-lb ton of bagasse is roughly equivalent to 6300 cu ft of 1000 Btu natural gas, to 40 gal of fuel oil, or to $^1/_3$ ton of coal, depending on efficiency of steam-generating equipment in each case. Therefore relative fuel costs can be approximated closely.

Because of its bulk, which averages 7 lb per cu ft as produced, it must be baled for storage. In Louisiana the bales weigh about 250 lb, measuring 17 in. × 21 in. × 30 in., or 6.2 cu ft. The baled bagasse, therefore, weighs roughly 40 lb per cu ft so that it occupies only $^1/_6$ as much space as the loose material.

Each baling press, which is essentially a reinforced hay baler, produces about 200 tons per 24-hr day. The baling stations are relatively expensive to build, operate, and maintain, so that in Louisiana it has been found impractical to contract for baled bagasse in quantities less than about 100,000 tons per year.

After baling, the bagasse is stored near the mill in loosely stacked piles, which are covered by sheet-metal plates. Aided by heat produced from initial fermentation, the bales dry out to about 25 per cent moisture, in several months' storage, after which period they can be stored for several years with little or no

[3] Engineering Department, Combustion Engineering, Inc., New York, N. Y.

further change, except for a loss of about 5 per cent due to ground moisture on the bottom layers.

The cost to the user in Louisiana is made up of the following factors:

Purchase price—fuel value plus a bonus, based on 16⅔ c/million Btu for equivalent gas fuel/ton bone dry (TBD)	$2.50
Baling, stacking, and covering in field (TBD)	6.00
Loading costs from storage to cars (TBD)	.50
Total cost at point of origin (TBD)	$9.00

To this must be added transportation costs to the user's plant, which in Louisiana averages about $2, so that bagasse cost totals about $11 per ton delivered. It is, therefore, not a cheap raw material and is feasible only when shipping distances are quite short, such as near producing areas, and a low-price alternate fuel is available, as is the case in Louisiana. It will be noted that 75 per cent of the cost to the user comprises baling, storage, and transportation.

In Louisiana about 200,000 tons of the 750,000 tons of bagasse produced per season are used for wallboard, the balance being retained as boiler fuel.

Costs in other localities, of course, will vary somewhat according to labor rates, fuel costs, shipping facilities, etc.

3 Furfural production from bagasse has been accomplished successfully, but according to recent published data, its cost up to the present has been higher than that of other suitable raw materials. A plant is now being built in Puerto Rico for this purpose.

The total amount of bagasse which would be required for this process is relatively small, as one or two factories in the West Indies could supply most or all of the furfural used in the United States.

Furfural is a colorless to reddish-brown liquid, soluble in water, alcohol, or ether, used in the manufacture of synthetic resins and molding compounds, disinfectants, deodorants, as a glue preservative, nitrocellulose solvent, and numerous chemical derivatives, and in connection with the refining of lubricating oils. Other raw materials used for furfural production include oat hulls, cottonseed hulls, corncobs, rice hulls, and so on.

The furfural residue may be processed into a cattle feed, or burned as fuel after dewatering.

4 Local production of wallboard, and the like, from surplus bagasse is being accomplished in some mills on a relatively small scale. Here again the matter of economics is of primary importance and the available bagasse surplus will depend on type of mill, efficiency of steam generation, whether raw or refined sugar is produced, and the fiber content of the cane. Wallboard is an excellent insulation against heat as well as cold, when properly treated to resist moisture, insects, etc., so that local markets will doubtless be developed as such uses become established.

The use of surplus bagasse for such purposes obviates the need for comparing its cost against that of an imported alternate fuel, as well as the cost of transportation of the raw material. In many cases the bagasse must still be baled, or dried by other means, which the foregoing tabulation shows to be major cost factors. Maximum efficiency in the steam-generating equipment is highly desirable for the situation outlined.

5 Flash or rotary-kiln drying of bagasse for direct utilization in paper, wallboard, or furfural production, without necessity for baling and storing, or for more efficient performance as a boiler fuel, has been investigated carefully and is practiced to a considerable extent in Louisiana, where the driers are natural-gas-fired using their low-cost local fuel.

The waste gases from the stack cannot be used economically for this purpose as the temperature is too low for the heat input required. For example, to reduce the moisture from 50 to 35 per cent, the following values have been determined: For a capacity of 1500 tons of bagasse per 24-hr day or 62.5 tons per hr, about 160,000 lb of gases at 1400 F are required. This is equivalent to 50,000,000 Btu/hr.

To dry this 62.5 tons of bagasse per hr would the reforerequire the heat available from 6 tons of bagasse, or 338 gal of oil, or the boiler output would be reduced by 30,000 lb per hr by removal of the hot gases, according to the foregoing. The fuel cost of this drying therefore can be closely approximated.

In addition to the fuel cost, the wet bagasse must be conveyed to and from the drier; gas ducts and fans must be provided, power, labor, and supervision considered, all of which factors increase the cost of such drying.

Experience to date has indicated that for boiler fuel the cheapest way to dry the bagasse is in the furnace, using boilers with stokers, waterwalls, and air heaters to secure maximum efficiency in steam generation.

Undoubtedly there are many by-product uses for bagasse which can be justified economically. However, careful studies are required as local conditions will largely determine whether the bagasse should be used solely as fuel, or in whole, or in part, for other purposes. Many sugar mills are located in isolated sections, away from other sources of fuel, while some can profit by their proximity to markets and fuel sources, to process their bagasse instead of burning it.

REFERENCES

1 "Louisiana Sugar Cane Bagasse," by A. G. Keller, Professor of Chemical Engineering, Louisiana State University, *Paper Trade Journal*, May 2, 1952.

2 "Burning Bagasse at Cuban Sugar Centrals," by H. G. Meissner, *Combustion*, January, 1948.

3 "Flash Drying Bagasse for By-Product Production," study by Combustion Engineering, Inc., for Cuban project, March, 1953.

4 "Furfural," editorial, *The Sugar Journal*, November, 1953.

ALEJANDRO TORT O.[4] The paper includes some of the writer's personal experiences in the burning of bagasse in several sugar mills of this country.

The author points out several of the advantages of the spreader-stoker type of bagasse-burning. The writer would like to refer to the preference of the technical and operating personnel for the spreader stoker, once they are familiar with this relatively new type of burning bagasse.

The operation of any bagasse furnace is influenced largely by the operation of the mill. In a well-operated plant the bagasse flows constantly from the grinding mill. Interruptions seldom occur. The capacity of the mill is set so as to grind the cane at a constant rate during long periods, thus producing a constant amount of bagasse. All the main sugar mills in this country refine their own sugar, and the combustion of the bagasse is insufficient to generate all the steam required. All the bagasse is burned and, in addition, some oil as auxiliary fuel. The operation is best carried out by burning the bagasse at a constant rate and taking the load swings with the auxiliary fuel. Seldom does the steam demand drop to a point where it is necessary to reduce the amount of bagasse fed to the boiler.

Once the milling operation is normalized, the speed of the bagasse feeders and the air flow to the spreader stoker can be set to correspond to the desired boiler output. The spreader stoker can run indefinitely without further adjustment until the bagasse flow is interrupted. The changes in steam demand are taken by the oil burners of the stoker-fired units or by separate oil-fired boilers. The type of furnace used with the spreader stoker per-

[4] Combustion Engineering de Mexico, S. A., Tercer Piso, Mexico 1, D. F.

mits a simple installation of oil burners of sufficient size to generate full steam capacity of the unit.

With hearth-type furnaces, there is no way to regulate closely the amount of bagasse fed to the boiler. The operator frequently has to adjust the bagasse and the air flow to the furnace.

The refuse must be removed from the hearth furnaces every shift. It is a difficult task to clean these units. There are large doors in the front of the furnaces provided for that purpose. When the fire has to be cleaned, the fuel and the air are shut off on one of the hearths. The door is opened and the operator agitates the bagasse fire so that it burns out as quickly as possible. When this is accomplished, the refuse is removed with long scrapers. Some of the refuse slags on the walls of the hearth and is broken loose with long bars; eventually this damages the brickwork. During all this time the operators are required to work in front of the open door and very close to the fire. A large amount of uncontrolled cold air is admitted, which chills the brickwork of the unit.

The spreader-stoker fire is cleaned in a very different way. The operator dumps the grates; then he takes his time, and when everything is normal again, he shuts off the air to one of the sections of the stoker. Then he opens the ashpit door and pulls the ashes out of the ashpit. The feeding of the bagasse to that section is not interrupted. The air admitted by natural draft through the ashpit door is used in the combustion of the bagasse fed to that section of the stoker. The remainder of the fire is cleaned in the same manner. The ashes are cold when removed and are in the form of a fine dust. They never slag or stick to the grates, even when preheated air is used.

To the sugar-plant engineer, as the author points out, this means that with boilers equipped with spreader stokers and a well-operated mill it is possible to run the boilers at the capacity desired around the clock, and as long as necessary, with less maintenance work. With the hearth type of furnaces, the losses of capacity during the cleaning period are inevitable and it is necessary to shut down the boilers frequently to clean out slag.

C. H. McAlister.[5] Bagasse burning is a subject very close to my heart and although I do not propose to know all the angles to the proper burning of it, I do have some very definite ideas about what one does not do in firing it.

We are presently erecting and will have in operation about August, 1954, a Combustion Engineering Type VU-4OS, high-set, single-pass, tangential and suspension combination bagasse and oil-burning steam generator with a capacity of 150,000 lb per hr with bagasse and 190,000 lb per hr on oil.

This unit will operate at 415 psig and 760 F and is equipped with a rotary grate.

In particular, we chose this type of boiler with this type of grate for a number of reasons, some of which are as follows:

1 Ash from Hawaiian cane is very obstinate; with considerable field dirt coming in with the bagasse, the tendency of slagging is great. Therefore any other type of grate might well plug up and shut the boiler unit down.

2 The single-pass unit, we believe, will permit less carry-over, hence less trouble in the air heater and less foreign material to settle around the countryside.

3 Higher pressure and temperature steam will give us more economy and less fuel-oil-burning during the grinding season and good efficiency in fuel-oil-burning during the off season.

We generate considerable electrical energy for outside irrigation use and for other than our own purposes, hence the desired efficiency.

[5] Factory Superintendent, Kekaha Sugar Company, Limited, Kekaha, Kauai, T. H.

The VU-4OS steam generator will be equipped with Bailey control instruments and is expected to be fully automatic. These controls will change the fuel from bagasse to oil and from oil to bagasse, or a portion of both, as steam demand requires when bagasse is not sufficient to supply the complete demand.

Bagasse will enter the generator at 16 ft above the grate and it is anticipated that approximately 85 per cent of the bagasse will be burned in suspension and 15 per cent on the grate. The carry-over will be reinjected into the furnace.

We expect to have complete data on this unit before the end of 1954.

L. H. Coykendall.[6] Mr. de Lorenzi has presented interesting and useful information in this paper which those responsible for modernizing and improving efficiency in the Mexican sugar-industry power plants can use to advantage. Good instrumentation, properly maintained by operators who know how to use them, and good control of the fuel feed rate are essential for spreader-stoker-firing. While these aids are not essential to good operation of the Ward furnace or "pile-burning" method, their use should result in better efficiency, if that is desired. It is interesting to note that the installation of feeders on the Ward furnace of No. 7 boiler at El Potrero resulted in uniform steam generation.

It is unfortunate that data were not presented on the emission of bagacillo from the stacks of spreader-fired units compared to that from Ward furnaces. The carry-over of cinders from coal-fired spreader stokers is notoriously high, and the writer expects that with the relatively lightweight bagacillo it would be excessive compared to pile burning. In the author's presentation of the test on No. 5 boiler at El Potrero the "unaccounted for" portion of the heat balance represents the carbon loss in the fly ash discharged to the atmosphere with the stack gas. This loss is measurable by methods approved by the ASME and ASTM test codes and one wonders why it was not measured. In the tests reported by Gilg[7] for a Ward furnace this loss ranged up to 1.8 per cent and the writer would expect it to be considerably higher on a spreader-fired unit.

Improvements have been made in the equipment for the pile-burning method of firing including furnace water cooling and either a dump-type hearth or traveling grate can be provided if desired. In this way, cleaning time is reduced or completely eliminated and the advantages of the pile-burning method of firing are retained. Among the advantages for this method of firing are the following:

1 Steam output is not seriously affected by fluctuations or short stoppages in bagasse feed.

2 Loss of ignition due to bagasse moisture content above 50 per cent is practically unknown.

3 Operation is simple.

4 Costly instruments and combustion controls are not required to obtain good operation with acceptable efficiency.

Author's Closure

The author wishes to express his appreciation and thanks to those who prepared written discussions or contributed questions from the floor.

Mr. Meissner's discussion outlines many of the factors responsible for development of newer bagasse-burning methods. More and more sugar mills are taking advantage of the improved economies to reduce quantity of purchased fuel needed for plant operation.

[6] Engineering Coordinator, The Babcock & Wilcox Co., New York, N. Y. Mem. ASME.

[7] "Utilizing Bagasse as Fuel," by F. X. Gilg, Trans. ASME, vol. 70, 1948, pp. 719–727.

The extent to which bagasse is able to satisfy heat requirements of the sugar-making process is clearly indicated. Bagasse, in some areas, also has many nonfuel uses. Thus a positive monetary value can be assigned to it and mill managements must carefully consider the economics involved in the use of a purchased replacement fuel.

Mr. Tort has had considerable experience in the operation of many different types of bagasse-fired units. His remarks, therefore, are not only factual but authoritative. He conducted the tests at El Potrero. They are accurate and representative of normal operation because all instruments and measuring devices were calibrated both before and after the 24-hour test run. This accuracy is further emphasized by the additional efficiencies reported for boilers Nos. 4 and 6.

Mr. Tort's interesting comments regarding possibility for regulating closely fuel supply to hearth-type furnaces serve to emphasize the positive control provided by drum feeders, used with spreader-firing. In addition, he cites the practical aspects of extended hearth-cleaning periods, absent with spreaders, which adversely affect performance as to both capacity and efficiency.

Mr. McAlister outlines some of the reasons for selecting the recently developed *turbulent suspension burning* for bagasse firing. While the steam-generating unit for his mill is not yet in operation, practically all principal features of this newer burning method are being successfully used at another Hawaiian installation.

Mr. Coykendall infers that instruments and controls are absolutely essential to spreader-stoker firing but are not required for good operation of Ward or other pile-burning methods. This is a fallacy because spreaders can also be easily operated without use of instrument or automatic controls. He does, however, in a back-handed way, say, ". . . their use should result in better efficiency, *if that is desired.*"

Bagasse, as used in modern mills, has a definite monetary value. Thus improvement in efficiency and sustained capacity may be used to justify economically the expenditures permissible for installation and maintenance costs of instruments and controls. (Reports from a number of mills, which include refineries, state that oil consumption, after installation of bagasse spreaders, was reduced as much as 40 per cent.)

Instruments, by all means, should form a part of every installation to safeguard property and human life. The days of operating plants without regard to these two factors are fast passing, even in the remotest parts of the world.

Mr. Coykendall says that cinder carry-over ". . . . from coal-fired spreader stokers is notoriously high" and, therefore, bagacillo carry-over should also be higher with spreader firing than with Ward furnaces. This is actually a matter of design. His experiences, no doubt, are pretty well confined to coal-fired applications which use exceedingly high grate heat-release rates, with resultant high carry-over, as contrasted with designs employing more conservative rates and much lower carry-over. The bagasse-burning units described employ moderate release rates. Furnaces are proportioned to provide low rising velocity of products of combustion, as well as low entrance velocity to boiler section; consequently, bagacillo *fall-out* is high, thereby accounting for the low carbon loss reported by Tort. Throat restriction as used with Ward, and other pile-burning designs, produce high gas velocity and the corresponding bagacillo *carry-over* is high, as substantiated by Gilg's reported test results.

To assume that the addition of either dump-type hearth or traveling grate will improve pile-burning is somewhat erroneous. It is the "fire thin and often" principle which is so effective in producing the superior results with spreader firing.

Water cooling, if carried down to hearth level, may reduce somewhat time and labor needed to break up and remove large fused clinkers, a normal by-product of pile-burning.

None of the improvements to pile-burning designs thus far projected or tried out have been able to assure operation which is as simple, efficient, controllable, constant, and interruption-free as that provided by spreader installations now in continuous commercial service.

Mr. J. Cabrera,[8] chairman of the session at which the author's paper was presented, called attention to the fact that actual yield of sugar for Mexico is approximately 2.2 tons per acre, instead of the 16-ton average assumed from our reference[2] source. Owing to this change, the land under cultivation for cane growing, as cited in the first paragraph, should be increased to approximately 380,000 acres.

[8] Member of the Mexican Government Committee for the Reorganization of the Mexican Sugar Industry. Mem. ASME.

British Naval Gas Turbines

By G. F. A. TREWBY,[1] BATH, ENGLAND

During the past ten years six different designs of naval gas turbine have been developed and tested under Admiralty contract and two additional designs have been brought directly from manufacturers for evaluation and testing. Taking each of these eight gas-turbine projects in turn, the paper gives a brief history of development and manufacture, describes the special features of the design, and gives details of the operating experience to date, both ashore and afloat. The lessons learned from this operating experience are discussed. The author concludes that the greatest single advantage that the gas turbine can offer for warship applications is its ability to pack more power in less weight and space than any other prime mover. To achieve this, naval gas turbines must be of the open-cycle type and constructed on lightweight lines. The various factors likely to influence the use of gas turbines for naval applications are mentioned and the future role of the gas turbine in the Royal Navy is discussed.

Introduction

IT was not until the pioneer work of Sir Frank Whittle had produced the first British aircraft jet engine in 1941, that serious consideration was given to the development of naval gas turbines in Great Britain. At this time the R. A. E. Farnborough were also actively engaged in gas-turbine development and their first design, manufactured by Metropolitan Vickers, ran in December, 1941. The idea of using a naval propulsion unit based on the aircraft type of gas turbine was first discussed between Metropolitan Vickers and members of the Engineer-in-Chief's Department of the Admiralty in 1942, and a contract for the construction of three complete "Gatric" engines was placed in August, 1943.

In the ten years since this contract was placed, six separate designs have been developed and tested under Admiralty contract and a further two designs have been bought direct from manufacturers for evaluation and testing. Several of these gas turbines represent considerable advances in naval engineering and merit individual treatment. However, the present paper aims only at presenting a general picture of the current state of British naval gas-turbine development, with a review of the operating experience to date and the lessons learned. In the conclusions the factors affecting the future role of this new prime mover in the Royal Navy are discussed.

The paper is divided into three main sections as follows: Propulsion machinery for warships; propulsion machinery for landing craft and ship's boats; auxiliary machinery.

Appendix A gives design and constructional details of the naval gas turbines described, while the cycle diagrams with operating pressures and temperatures are shown in Appendix B.

[1] Commander (E), Royal Navy, Engineer-in-Chief's Department, Admiralty.

Sponsored by the Power and Fuels Division of the ASME Metropolitan Section and presented at a joint meeting with The Society of Naval Architects and Marine Engineers, on April 27, 1954, in New York, N. Y. Also presented at a meeting of the Power Division, Washington, D. C., Section in Washington, D. C., on April 28, 1954.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 13, 1954.

1 PROPULSION MACHINERY FOR WARSHIPS

One of the most obvious naval gas-turbine applications is for the propulsion of high-speed craft where the characteristics of this new prime mover can be used directly to obtain higher speeds. In many ways this is an ideal application and, being the most straightforward (as far as propulsion machinery is concerned), the progress in this field has been more rapid than elsewhere. Two different designs of simple gas turbine, the "Gatric" and "G.2" already have been operated at sea in British naval coastal craft.

"Gatric" Gas Turbine (1–6)[2,3]

Introduction

A contract for three simple-cycle gas turbines, based on the F2 aircraft jet engine but with an output turbine fitted in the tail pipe, was placed with Messrs. Metropolitan Vickers in August, 1943. The main object of the contract was to gain experience with the only type of gas turbine then available in Britain, a short-life aircraft unit. The project was given the code word Gatric and a sectional arrangement of the engine is shown in Fig. 1.

The F2 ran on kerosene and as this fuel was not considered suitable for naval applications the first part of the work consisted in developing the combustion system of the engine to burn diesel oil. It was found that very few modifications were necessary. Carbon formation was experienced at first, but with minor alterations to the fuel jets and the size and position of the holes for the admission of primary and secondary air, together with a slight increase in diameter of the annular combustion chamber, satisfactory combustion was achieved.

Bench tests on the first complete engine began in April, 1946, and some alteration to the capacity of the compressor turbine was found necessary before the designed performance could be achieved. This alteration brought the turbine working line too close to the compressor surge line at low powers and blowoff valves at the fifth compressor stage were necessary to facilitate starting. The specific fuel consumption of the Gatric at various powers obtained during short testing is shown in Fig. 2, together with the performance curves of other naval propulsion gas turbines.

Installation of the first engine in MGB.2009 (later renumbered 5559) was completed in July, 1947, and in August of that year MGB.2009 made history by being the first vessel in the world to be propelled at sea by a gas turbine, Fig. 3, just 50 years after the trials of the *Turbinia*. The main objectives of this development were to gain experience of installing, operating, and maintaining a gas turbine at sea. Details of the arrangement in MGB.2009, in which Gatric replaced the center of three Packard gasoline engines, are shown in Figs. 4 and 5 and the arrangement of air-inlet ducting to the compressor and the air circulation past the gas turbine are clearly visible.

During the early sea trials trouble was experienced with bad temperature distribution at the outlet from the combustion chamber. This resulted in distortion of the combustion chamber and overheating of the first-stage power turbine blades. By this time Messrs. Metropolitan Vickers had successfully developed a film-cooled chamber and when this new design was incorporated in the engine of MGB.2009, no further trouble was experienced.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

[3] Appendix A, column 1; Appendix B, diagram 1.

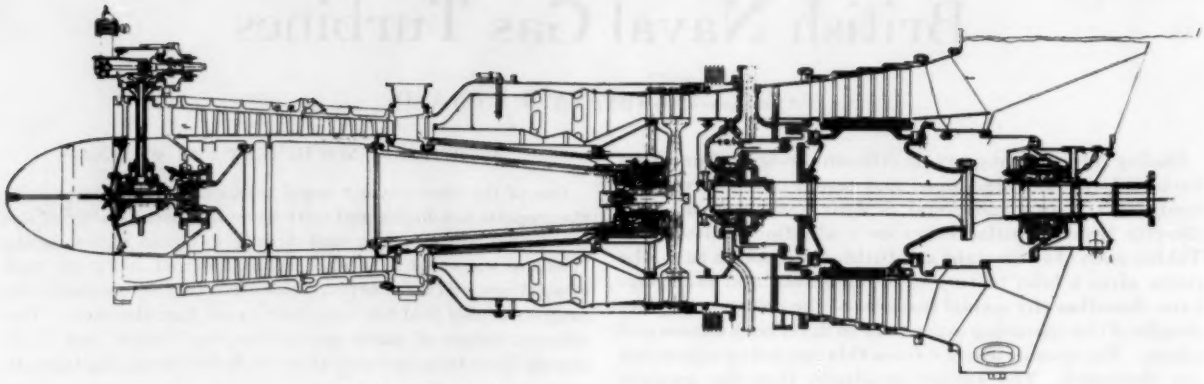

FIG. 1 SECTIONAL ARRANGEMENT OF GATRIC GAS TURBINE

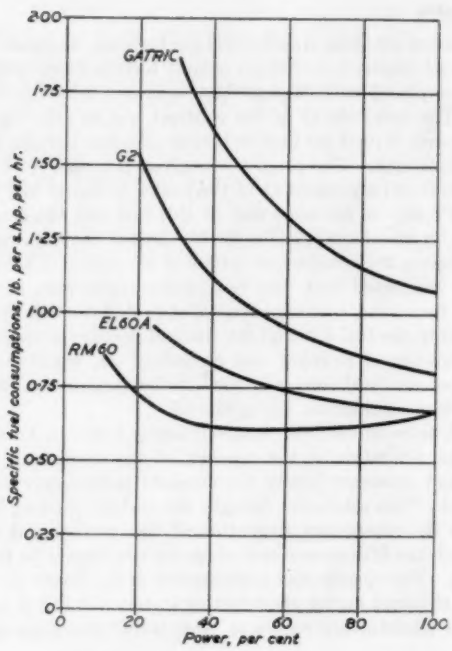

FIG. 2 SPECIFIC FUEL CONSUMPTION OF PROPULSION GAS TURBINES. ACTUAL TEST RESULTS USING ADMIRALTY DIESEL FUEL (GAS OIL), CALORIFIC VALUE 18,000 BTU/LB. (10,300 CHU/LB)

FIG. 3 MGB.2009 IN SOLENT, AUGUST, 1947
(Courtesy, "The Sunday Graphic.")

It did not, however, completely eliminate stratification and some 50 deg C variation in the readings on the four thermocouples fitted at the power-turbine inlet remained.

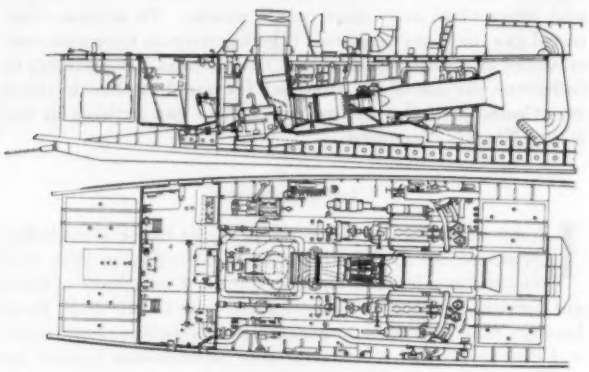

FIG. 4 ELEVATION AND PLAN SHOWING DETAILS OF GATRIC INSTALLATION IN MGB.2009

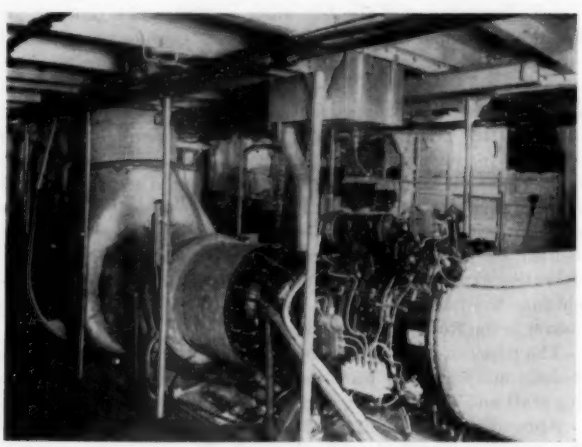

FIG. 5 GATRIC GAS TURBINE AS INSTALLED IN MGB.2009

Lessons Learned From Gatric Trials

Sea trials of the Gatric gas turbines in MGB.2009 continued over a period of four years and were remarkably trouble-free. At the end of this time the following points were established:

(*a*) *General.* The operation of a simple gas turbine at sea was a practical proposition which presented no insuperable difficulties and the general characteristics of this new prime mover were particularly well-suited to the propulsion of high-speed coastal craft.

(*b*) *Compressor Fouling.* The deterioration in gas-turbine performance caused by compressor fouling is well known and it was anticipated that the salt-laden atmosphere might affect the performance of naval gas turbines, particularly in a comparatively small high-speed vessel such as MGB.2009. The air-intake arrangements shown in Fig. 4 were therefore designed to prevent salt spray entering the compressor, air being drawn from a sheltered position abaft the bridge and passed through a settling tank before reaching the compressor inlet. This arrangement was not entirely effective, however, and after 120 hr running, the compressor efficiency had fallen to 95 per cent and the engine output to 86 per cent of their design values. Examination showed that the deterioration was caused by salt deposit on the compressor blades.

To overcome this fouling problem a water-spray ring containing five equally spaced nozzles was fitted in the compressor inlet. A 10-gal injection of distilled water with the gas turbine running increased compressor efficiency and output to 98 and 97 per cent of design and a second injection restored them to their design values. A routine was then established in which 10 gal of water were injected at the rate of about 2 gpm every 3 to 12 hr, depending upon the drop in engine performance. This simple procedure was completely successful in combating the effects of compressor fouling both from salt spray and fog. Similar injection equipment has been fitted to the later G.2 and RM.60 gas turbines.

(*c*) *Bearings.* Two bearing failures were experienced near the completion of the sea trials of Gatric and it began to appear doubtful if ball-and-roller bearings were capable of withstanding the high speeds and temperatures associated with gas turbines for long periods, particularly under seagoing conditions. In neither case was the cause of failure established.

(*d*) *Noise Levels.* Before any silencing action was taken a noise survey was carried out and it was found that the greatest noise level in the engine room 117 decibels (db) originated from the gas-turbine gearbox. On deck, just aft of the bridge, the level was also 117 db, the major source being traced to the compressor-air intake; on the bridge itself noise from the funnel predominated, the total level being 102 db.

The most serious component of noise was considered to be the high-frequency note emanating from the axial compressor and transmitted through the air-intake trunking and settling chamber. By lining the settling chamber with fiberglas and fitting "splitters" of the same material in the air intake, a reduction in the compressor-blade note of 39 db was achieved. This reduction made possible a more accurate evaluation of the exhaust noise and it was found that the compressor-blade note was present there also. Silencing equipment including a "torpedo-type" splitter was fitted in the funnel and was most effective in reducing funnel noise.

(*e*) *Compartment Cooling and Heat Insulation.* Owing to the proximity of two Packard gasoline engines, great care was necessary in the design of the Gatric's heat insulation and it was decided to enclose the entire engine in a ventilated casing (see Fig. 4). The gas turbine itself was lagged with 2-in. asbestos mattresses and a light metal casing was fitted over this lagging but not in contact with it. Air was drawn between the casing and lagging by an engine exhaust operated ejector in the funnel. This method of cooling the gas turbine and the compartment by means of air-flow insulation was extremely effective and the principle has been used in later installations. (A 12½-in. fan was originally fitted to supply air to the engine casing but the exhaust ejector in the funnel proved so efficient that the fan was removed).

(*f*) *Compressor-Blade Material.* The original Gatric compressor blades were of R.R. 56 anodized by the chromic-acid process. Examination after the first 50 hr running revealed that the blades had suffered local intercrystalline corrosion, the average depth being 0.005 to 0.01 in. The blade material was changed to R.R. 57 and no further corrosion was experienced although it might still occur in the absence of water-washing.

(*g*) *Loss Due to Trailing.* Initially Gatric was fitted with a clutch located between the propeller and reduction gear (and integral with the gear) to prevent the free-power turbine from trailing when the boat was powered only by the gasoline engines. Trials indicated that this clutch was unnecessary, however, since the power lost in "windmilling" the turbine with both gasoline engines developing full power was only 1.3 per cent corresponding to a boat speed of 0.18 knot.

Residual Fuel Tests on Gatric (7, 8)

Concurrently with the sea trials, testing was carried out with the other two Gatrics at Metropolitan Vickers works and the Admiralty Engineering Laboratory. The engines were used for proving new components and for trials with various types of fuels including residual boiler fuels. The residual-fuel tests included a continuous run of 48 hr at maximum cruising load and showed that the engine was capable of running for limited periods on residual fuel oil to Admiralty standard specification. The materials of construction stood up well with the exception of the fuel-jet shields and, on the whole, the engine was less adversely affected than anticipated.

A total of 200 hr running on residual fuel was completed at the Admiralty Engineering Laboratory and from the appearance of the engine after the tests, it was predicted that 1000 hr could have been completed without unacceptable loss of performance or failure of the turbine-blade materials.

As a result of these trials, equipment for burning residual fuel, including an exhaust-operated fuel-oil heater, was fitted in MGB.-2009 but only one short run could be completed before the hull was condemned and the vessel scrapped.

The principal lessons learned from the residual-fuel trials with Gatric engines were:

(*a*) That a high degree of filtration of the fuel was necessary.

(*b*) That it is preferable to pump the oil cold if wear of the pumps is to be kept to a minimum.

(*c*) That exhaust heating of the fuel is a practical proposition.

(*d*) That a simple Naval gas turbine operating with a maximum gas temperature of 750 C (1382 F) can burn a residual fuel oil for limited periods.

Gatric Life

It should be noted that although Gatric was designed for a life of only 300 hr, two of the engines have now operated for nearly 600 hr and this brings home the important point that when the "life" of a gas turbine has expired, it means only that certain components require renewal. It does not involve scrapping the whole engine as was originally supposed in some quarters.

G. 2 Gas Turbine (9, 10)[4]

Introduction

The Gatric project was originally planned as a convenient floating test bed for the only kind of gas turbine then readily available—a short-life converted aircraft unit. But it soon became apparent that this type of engine had many advantages for the propulsion of high-speed craft and consideration was given to further installations on these lines.

In December, 1948, a contract was placed with Messrs. Metropolitan Vickers for the design and manufacture of four larger gas turbines based on the Beryl aircraft jet engine, but with the ad-

[4] Appendix A, column 2; Appendix B, diagram 2.

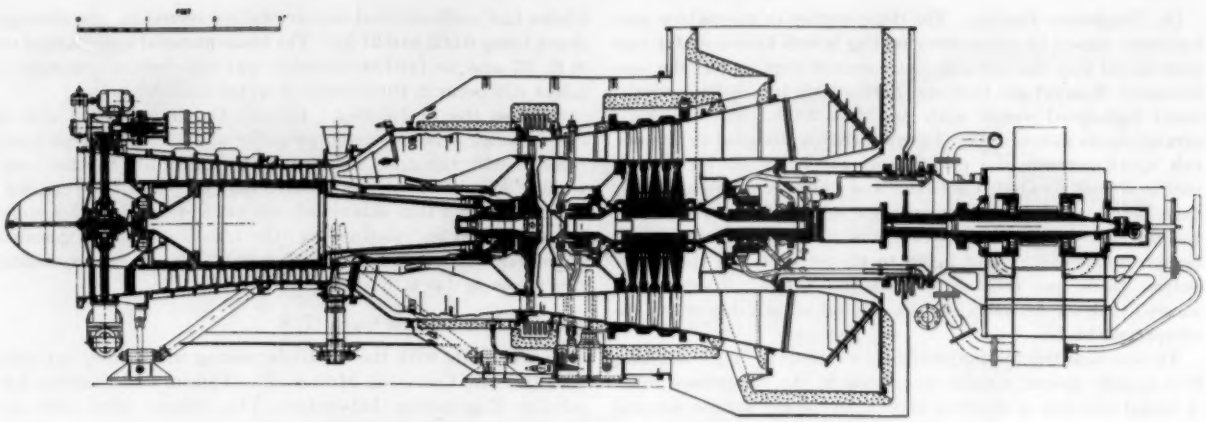

FIG. 6 SECTIONAL ARRANGEMENT OF G.2 GAS TURBINE

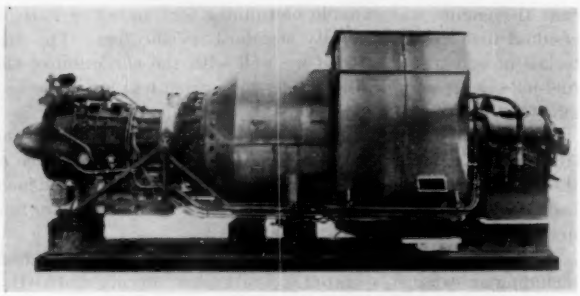

FIG. 7 G.2 GAS TURBINE ON TEST STAND

dition of a separate power turbine driving the output shaft through single helical reduction gearing. This design, known as the G.2, was ordered for installation in two fast patrol boats, *Bold Pioneer* and *Bold Pathfinder*. The gas turbines were intended for use at high speed only, diesel engines being fitted for maneuvering and cruising. Each engine drives a separate shaft, the two gas turbines being on the wing shafts and the diesel engines on the center shafts. Fig. 6 shows a sectional arrangement of the G.2 gas turbine and a view of the engine in the maker's works is shown in Fig. 7.

Shore Testing

Manufacture of the first G.2 was completed in 1951, but initial testing revealed that the power turbine was not correctly matched with the gas generator, the maximum power available at the designed gas temperature being 3800 hp. Since manufacture of modified power-turbine nozzles would have involved considerable delay, it was decided to install the first two G.2's at the reduced rating in order to obtain early sea experience.

The third engine with modifications to improve matching went on test at the firm's works in December, 1951, and the specific fuel consumption obtained over the whole power range is given in Fig. 2. It will be seen from this figure and Appendix A that the G.2 gas turbine has a much improved fuel consumption compared with the Gatric, a lower specific weight, and a longer life. This great improvement in performance achieved over a period of five years is most encouraging and there is every indication that the trend will be continued in future designs. During shop trials noise measurements were taken and it was found that the noise level beside the engine was 102 decibels under idling conditions, rising to 114 decibels at full power.

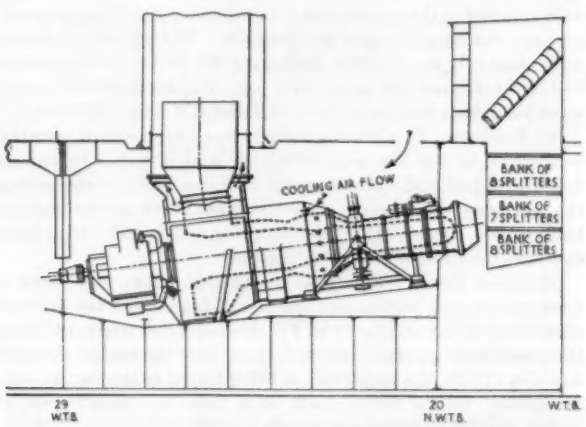

FIG. 8 DIAGRAM SHOWING INSTALLATION OF G.2 GAS TURBINE IN "BOLD BOATS"

FIG. 9 HMS "BOLD PIONEER" AT SPEED
(Courtesy, "The Times.")

Installation Arrangements in "Bold" Boats

A diagram illustrating the installation arrangements of the G.2 gas turbines in the Bold boats is shown in Fig. 8, while Fig. 9 is a view showing *Bold Pioneer* at speed. The air intake and air-flow-cooling arrangements are similar to those employed with Gatric in MGB.2009, the ejector effect of the exhaust gases being used to draw air into the engine compartment and over the hot parts of the gas turbine, thereby providing an air-flow insulation

system considerably lighter than conventional lagging. No clutch is provided, the gas turbines being connected directly to the propeller shafts through a reduction gear of 4.73 to 1. Starting is by means of a small swashplate air motor which can be seen on the top left of the compressor in Fig. 7. Gatric had been started very successfully by an electric motor but as an air supply was necessary in the Bold boats for starting the diesel engines it was decided to adopt this method for the gas turbines also.

Trials at Sea

Sea trials of the G.2 gas turbines in the Bold boats began toward the end of 1952, and many more problems have been encountered than in the original Gatric installation.

The first was inadequate gearbox scavenge-pump capacity. Originally, two 35-gpm scavenge pumps and a single 35-gpm pressure pump were fitted but it was found that the capacity of the two scavenge pumps operating in parallel from a single suction pump was not adequate and flooding of the gearbox resulted. The trouble was overcome by fitting a single 45-gpm scavenge sump and restricting the capacity of the pressure pump to 25 gpm.

After some 50 hr operation in *Bold Pathfinder*, repeated stalling of the starboard gas turbine occurred at gas-generator speeds of about 5500 rpm. There was no apparent loss of power or excessive vibration and the gas temperatures were normal. As the compressor approached the stalling speed a slight internal rumbling sound could be heard which suggested the onset of compressor surge. On opening up the compressor it was found that six blades from the last moving row were broken near the root and all bore the characteristics of fatigue failure. Some of the blades had passed through the combustion chamber and had satisfactorily aluminized the compressor turbine without any apparent damage!

It was known that the natural frequency of the last compressor-blade row coincided with a possible excitation from the 10 outlet struts at 5500 rpm, but it was thought that the two rows of fixed guide vanes would prevent excitation from this source. Closer examination of the design, however, indicated that the proximity of the struts to the last row of guide vanes and the aerodynamic shape of the passages between the unstaggered struts would aggravate rather than dissipate an excitation from the outlet struts. Fig. 10, showing an expanded plan view of the last rows of compressor blading and the outlet struts as originally fitted, illustrates this point. The defect was cured by removing the second row of outlet guide vanes and modifying the first row to suit, as shown in Fig. 11. The space left between the guide vanes and the outlet struts was then sufficient to dissipate any upstream disturbances and no further trouble has been experienced from this source.

As an additional precaution, automatic air blowoff valves have been fitted which are open on starting and close at a gas generator speed of 6500 rpm. These minimize the risk of surging which would greatly increase any excitations set up from the outlet struts.

No sooner had this defect been successfully overcome than another vibration problem arose, first manifesting itself in the starboard gas turbine of *Bold Pioneer*. A first-row compressor moving blade failed causing considerable consequential damage. This particular blade had been noted as slightly damaged when the compressor was opened up after the acceptance trial, presumably by the passage of some foreign body through the compressor. This fact tended to obscure the real cause of failure but was a timely reminder of the very severe damage which can arise if any loose gear, etc., gains access to the intake ducting of gas-turbine installations.

About five months later some second-stage fixed and moving blades of *Bold Pathfinder's* starboard gas turbine failed. A

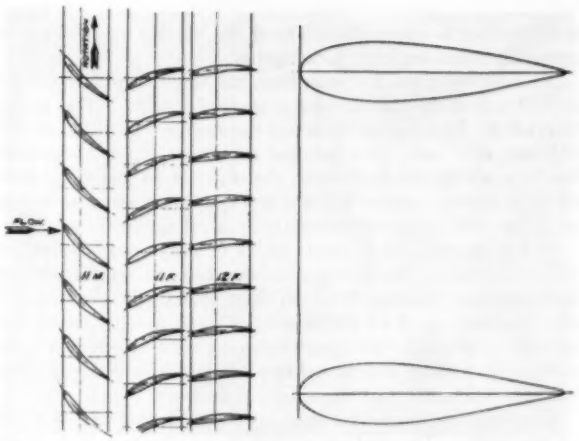

FIG. 10 EXPANDED PLAN VIEW SHOWING DETAILS OF G.2 COMPRESSOR BLADING AT OUTLET END—BEFORE MODIFICATION

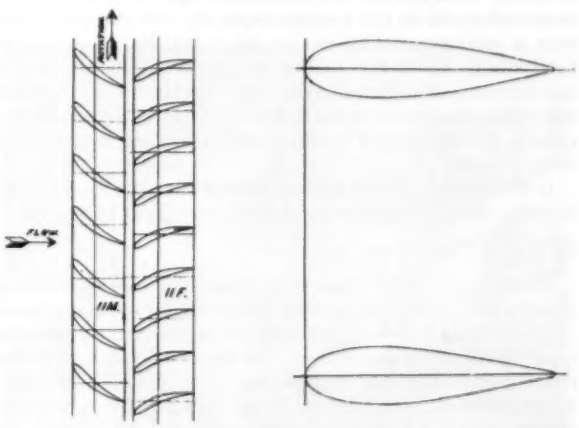

FIG. 11 SAME VIEW AS IN FIG. 10—AFTER MODIFICATION

thorough investigation was then made into the cause of failure including the fitting of strain gages on compressor blades to determine the conditions at which excessive amplitudes occurred. As a result, it was definitely established that both failures had been caused by "rotating stall." This is an aerodynamical effect which is liable to occur when the L-P stages of axial compressors go into stall at low speeds. (A brief explanation of this phenomenon is given in the footnote.[5])

The blade-vibration tests showed the existence of a group of gas-generator speeds between 3000 and 4800 rpm at which resonant vibration was likely to occur. This is below the normal propulsion range, but includes the speed at which the engines have been run after periods at high power to cool the bearings prior to shutdown. The gas-generator speed for idling and cooling down has been temporarily raised to 5000 rpm and design

[5] *Note on Rotating Stall.* The explanation of this phenomenon is briefly as follows: When a compressor-blade row stalls it does so initially in patches, which tend to block the flow, and the air therefore diverges to each side of the patch. This has the effect of unstalling the blades in the row ahead of the patch and stalling those behind. The result is a rotation of the stalled patch in the direction of blade rotation but at a reduced speed. This rotation of the stalled patches excites the blades in the fixed and moving rows and they are liable to vibrate in resonance at excitation frequencies depending upon the number and rotational speed of the stalled patches, both of which can vary.

modifications to insure the safety of the blading over the whole operating range are being investigated.

An interesting point arising from the three blade failures described is that the G.2 compressor is closely similar to that of the Beryl Mark I jet engine which had successfully completed an Air Ministry type test. The fact that an aircraft-engine compressor has been cleared for flight duty, therefore, is no guarantee that blade-vibration troubles will not be experienced when operating at low speeds, as may be necessary in naval applications.

One of the important points which it was hoped to establish from the trials of the G.2 gas turbines was the behavior of the ball-and-roller bearings fitted to these engines. Unfortunately, the experience in *Bold Pathfinder* has been masked by an extremely severe hull vibration which occurs under certain conditions when turning and caused two failures of the power-turbine bearings during the early operation of the boat.

Investigations were made into the hull vibration and its effect on the engine. The analysis showed that a vibratory load, much greater than the normal thrust load, was occurring on the ball bearings. This caused the balls and cage to vibrate axially between the races and led to failure of the cage. Such vibration is most undesirable in ball bearings especially with the air-oil mist type of lubrication which was employed initially. A change to flood lubrication of this bearing was made to improve the oil supply both under normal operation and when trailing. At the same time, limitations on the operation of the boat have been imposed to prevent the hull vibration and no further bearing failures have occurred.

In *Bold Pioneer*, which has not suffered from this trouble, presumably due to the different hull form, no bearings have failed.

G2 General Conclusions

There is no doubt that there have been serious teething troubles during the initial trials of the G.2 gas turbine at sea, many of which would have been avoided if the engines had been given a more thorough testing ashore. The successful and trouble-free running of Gatric had perhaps made the Admiralty overconfident, and the first G.2 was run for an aggregate of only 37 hr before carrying out the official acceptance test of 9 hr prior to installation.

Nevertheless, the difficulties experienced during the initial trials were not inherent in the principle of the gas turbine and the G.2 has again shown that, basically, the characteristics of the simple gas turbine are ideal for the propulsion of high-speed coastal craft. Other lessons learned to date from the sea trials of the G.2 are:

(*a*) *Starting and Flexibility of Operation.* Air starting has proved quick and reliable, the time taken to reach idling speed being about 30 sec under normal conditions. The rate of acceleration thereafter is servocontrolled and, although the throttle lever can be moved "instantaneously" from idling to full power, the fuel flow is gradually increased over a period of about 12 sec, corresponding to the maximum surge-free rate of acceleration.

The automatic blowoff valves fitted to the compressor to avoid surge at low powers or when rapid changes of speed occur have been successful and trouble-free in operation.

(*b*) *Compressor Fouling.* Water-spray arrangements similar to those installed in MGB.2009 were fitted to the G.2's, air being tapped from the compressor and used to blow distilled water into the spray ring fitted at the compressor inlet. This simple method has again proved extremely successful in restoring performance after salt-spray fouling. The usual practice is to wash the compressors through daily at 5000 rpm using 10 gal of distilled water over a period of about 3 min. This frequency is not normally necessary from the performance point of view but is considered desirable in order to prevent corrosion.

(*c*) *Cooling Down.* If the gas turbines have been operating at high power the gas generators are run for 5 to 15 min at 4750 rpm to cool the bearings down to a temperature of about 90 C (194 F) before shutting down. On occasions, however, the engines have been shut down immediately after high power running without consequent damage.

(*d*) *Instrumentation.* Accurate recording of gas temperature is essential for the proper control of naval gas turbines. Several failures of gas-temperature-recording instruments have taken place in the Bold boats and false readings have occurred due to hull vibration at high speed. It may be advisable in future designs to record gas temperatures at the turbine outlet rather than at entry to the power turbine as in the G.2's. This will enable a more robust instrument (designed for lower temperature) to be fitted. It also will safeguard the turbines as any internal failure will immediately result in a higher temperature at the end of the turbine-blade path.

Use of Simple Gas Turbines as Boost Units in Major Warships (11, 12)

Major warships operate for only very short periods at high power and it is wasteful of weight and space to carry around heavy long-life machinery whose full capacity is seldom used. A promising application for the gas turbine in naval vessels is therefore as a lightweight, short-life "boost" engine for use at high powers, in conjunction with a base load or cruising engine. This conception of using simple, short-life, lightweight gas turbines as "booster" units with more efficient longer-life machinery for cruising is an important principle which avoids the need to install unduly heavy machinery solely to cover a range of powers which is seldom used in warships. The gas turbine is the only prime mover which can take full advantage of the short life required in booster units to reduce weights and space without increasing maintenance difficulties.

In MGB.2009 and the two Bold boats the gas turbines and cruising engines drive onto separate shafts and piston engines are employed for cruising, but the boost principle is not necessarily confined to arrangements of this kind. Steam turbines, additional gas turbines, or free-piston gas generators can be used for cruising and the boost and cruising engines can be coupled to the same shaft if desired.

The characteristics required of boost gas turbines are extreme lightness and compactness, reliability, and reasonable efficiency at high powers with minimum air requirements. The life can be very short, judged by normal standards. These qualities are similar to those of aircraft gas turbines and in this field the British Navy is therefore in the happy position of benefiting directly from the vast aircraft-engine development program financed by the Ministry of Supply.

The development of naval gas turbines for main propulsion machinery presents a more difficult problem than the boost application. Nevertheless, the gas turbine may offer sufficient advantages, particularly in the power range between 5000 and 10,000 hp, to warrant developments of this type. Two naval gas turbines in this category have been built and tested under the code names of EL.60A and RM.60.

EL.60A Main Propulsion Gas Turbine (13,14)[6]

Introduction

A contract was placed with the English Electric Company, Ltd., Rugby, in September, 1946, for a gas-turbine plant (known as EL.60A) suitable for marine propulsion. The plant was intended to replace one set of steam machinery in the frigate, HMS *Hotham*, a Lend/Lease vessel which the United States

[6] Appendix A, column 3; appendix B, diagram 3.

kindly allowed the Admiralty to retain after the war for this special purpose. The *Hotham* was originally fitted with steam turboelectric machinery developing 6000 shp on each of two shafts, and appeared particularly suitable for a gas-turbine installation since reversing was already available, the power and alternator speed were reasonable, and a direct comparison with steam machinery could be made.

The main object of the contract was to obtain sea experience with a long-life gas-turbine plant at an early date. It was therefore decided to base the design as far as possible on existing knowledge and techniques. A lightweight set of advanced design was thus out of the question and, in many ways, it was desirable that the weight of gas-turbine plant should be similar to that of the steam machinery it was to replace. The EL.60A was designed to drive the *Hotham*'s starboard propulsion alternator at its original design speed and power, and the layout was adapted to fit in the existing forward engine and boiler room.

Cycle

The cycle adopted was a simple cycle with heat exchange, the compressor (or charging) turbine being in parallel with a geometrically similar but smaller power turbine driving the alternator. The maximum gas temperature at inlet to the turbine was 704 C (1300 F) and the heat exchanger was designed for 75 per cent heat recovery. Fig. 12 shows a model of the complete EL.60A gas turbine with details of ducting, starting motor, and main alternator, etc., while photographs of the actual set in the manufacturer's test house are shown in Figs. 13 and 14.

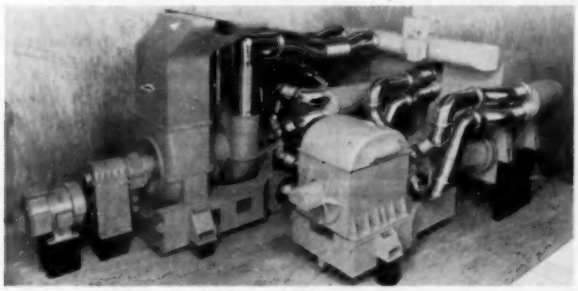

FIG. 12 MODEL OF EL.60A MAIN PROPULSION GAS TURBINE

Adoption of a parallel-flow cycle solved the problem of providing an efficient 6500-hp power turbine running at the alternator synchronous speed of 5600 rpm since only one third of the total mass flow passed through the power turbine. This cycle was also felt to meet the requirements for rapid manufacture, and the pressure ratio necessary for efficient operation was within the range of operating experience for axial-flow compressors. Maneuvering was also attractive since any power from full to zero could be obtained by manipulating the throttle and blowoff valves without altering the speed, mass flow, or pressure ratio of the compressor.

Control

Automatic control of the parallel-flow cycle of EL.60A necessitated a complex system, the basic principles of which were as follows: Normal control was by means of a single lever regulating the speed setting of the power-turbine governor. A movement of this lever in the "increase speed" direction started a small motor which increased the fuel supply to all combustion chambers at a predetermined rate until the power turbine reached the new governor-speed setting.

A movement of the lever in the "decrease speed" direction had an immediate and a long-term effect. The immediate effect was to reduce fuel supply to the power-turbine combustion chambers and, if the required speed change was appreciable, to close the power-turbine throttle valve and open a blowoff valve to atmosphere until the desired power-turbine speed was reached. Rapid change of output-shaft speed could thus be effected without immediately disturbing the fuel setting of the charging set.

The longer-term effect was a slow reduction in the fuel rate of the charging set and a consequent reduction in its speed. During this progressive reduction, the power-turbine speed was held at its governed value by a progressive opening of the throttle valve and

FIG. 13 EL.60A GAS TURBINE IN TEST HOUSE. VIEW FROM HEAT-EXCHANGER END
(Showing back of heat exchanger and No. 2 F.O. pp. Also side view of C.T. and bedplate cooling fan.)

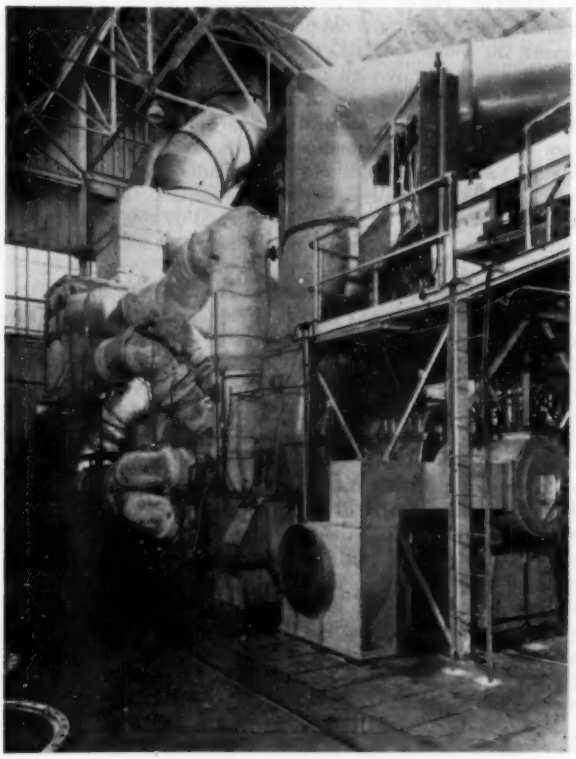

FIG. 14 VIEW FROM STARTER MOTOR END

a proportionate closing of the blowoff valve. Ultimately the set was restored to the normal condition of unthrottled operation at the desired speed, with the blowoff closed. If necessary, the long-term effect could be cut out and the charging set allowed to run continuously at a chosen speed; this condition met maneuvering requirements, as it enabled the time from "stop" to "full power" on the power turbine to be reduced to a few seconds.

Starting

The 250-hp starting motor was capable of rotating the charging set at 1000 rpm without fuel, but the normal procedure was to light the combustion chambers at 300 rpm and declutch the starting motor at 2000 rpm. On a cold start the set would become self-driving in five min and could sustain full load about 15 min later.

Manufacture of Set

It was originally planned to complete the manufacture of the set by the end of 1948. However, as work proceeded, it became apparent that a gas turbine of this nature could not be built using existing practice. Many of the manufacturing difficulties required the use of special techniques, some of which had to be developed or adapted especially for the job. All this caused considerable delay and it was not until 1949 that the first main component, the compressor, was ready for testing. At the designed operating point an over-all adiabatic efficiency of 86.2 per cent was obtained which was considerably higher than expected. The mass flow was also in excess of the design value and modifications were necessary to insure matching of the turbines. To achieve this the first stage of rotor and stator blades was removed from both turbines, thus increasing their capacities. It was hoped that any decrease in cycle efficiency caused by this modification would be offset by the increase in compressor efficiency compared with the design figure.

Further difficulties were encountered with the welding of the turbine rotors and it was not until September, 1951, that the complete set was finally ready for shore trials. Thus the original object of obtaining early sea experience had not been achieved while the initial trials of the RM.60 had already indicated the potentialities of a lightweight gas turbine for warship propulsion. The heavier EL.60A based on steam-turbine practice had, in fact, become obsolete and it was realized that the set would never become a prototype for future naval gas-turbine machinery. It was therefore decided to abandon the sea trials of the EL.60A in the *Hotham* and to terminate the project at the conclusion of the first test runs ashore. The performance obtained on these initial runs is shown in Fig. 2 but it is only fair to point out that several significant and unnecessary losses were located on test and with further development these could have been eliminated and the consumption figures thereby substantially improved. On completion of the test program in 1952 the set was stripped for a thorough examination of every component.

Lessons Learned

Although the original object of this gas-turbine project was not achieved and the development was subsequently overtaken by more advanced designs, much valuable information was obtained, the more important points being as follows:

(*a*) *General.* The cycle used, designed physically and mechanically on steam-turbine lines, is intrinsically heavy. Even with complete redesign, it is doubtful if the weight could ever be reduced below about 50 tons or 17 lb/hp.

Bearing in mind the weight of the moving parts and that the turbine casings are horizontally split and uncooled, the set appeared to be sufficiently flexible in operation for naval use except that in the event of a surge it was necessary to slow the charging set down below the self-driving point before the plant could be put on substantial load again, and this operation took five to ten minutes.

(*b*) *Compressor.* A very sound compressor, both aerodynamically and mechanically, has been developed, and the design could be confidently used as a basis for future projects.

It has been realized that a high efficiency over a wide range (over 85 per cent from pressure ratios of 1.5 to 4.2 obtained with this compressor) is often associated with a poor low-speed surge line. Thus more efficient over-all performance may result from improving the surge line by deliberate interstage blowoff at low-speed operation rather than by attempting to improve the complete compressor surge line by blading modifications.

(*c*) *Heat Exchanger.* The method of securing the tubes in the tube plates by induction brazing has proved satisfactory.

The aluminum-bronze tubes have been corroded during the short duration of the tests and, so far as can be judged from this limited experience, the material appears unsuitable.

Correct spacing of the tubes between tube sheets has been satisfactorily achieved by means of spiral wire spacers.

(*d*) *Combustion Chambers.* The louvered can-type combustion chambers developed by the Shell Petroleum Company for the EL.60A were the first of their type. They were most successful, producing perfect combustion without any trace of smoke above a preheat temperature of 300 C (572 F), and remaining completely free from any distortion or corrosion throughout the whole of the tests. Details of the chamber are shown in Fig. 15 and it will be observed that the louvered wall construction is achieved by packing together large members of truncated conical rings.

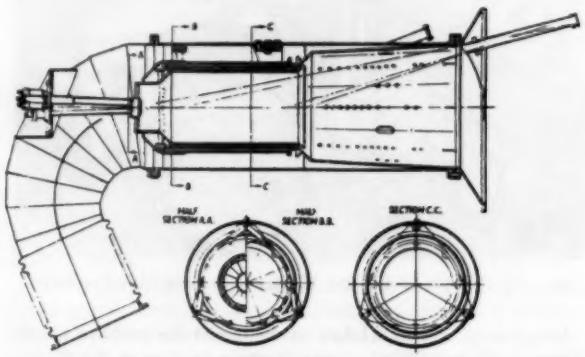

FIG. 15 DETAILS OF LOUVERED COMBUSTION CHAMBER FITTED IN EL.60A

Difficulties were experienced in matching the outputs of the multiple combustion chambers. (Unless this can be achieved the turbines are liable to be locally overheated.) The matching was particularly difficult at low powers where the individual burner output was very sensitive to minor manufacturing errors, burrs, etc.

(*e*) *Turbines.* The turbine-rotor and blade-root cooling system (details of which are shown in Fig. 16) proved extremely effective, although complicated to construct. Air tapped from the compressor delivery was, after filtration, fed into the center of the rotors. The air then passed outward to the blade roots and the spaces beneath the segmented disk covers around the rotor periphery. By this means the rotor disks and the more heavily stressed parts of the blades were protected from direct impingement of hot gases. Fusible plugs indicated that the cooling system maintained the rotor everywhere at least 250 deg C (482 deg F) below the hot-gas temperature. This highly effective cooling system, which required somewhat less than 2 per cent of the total

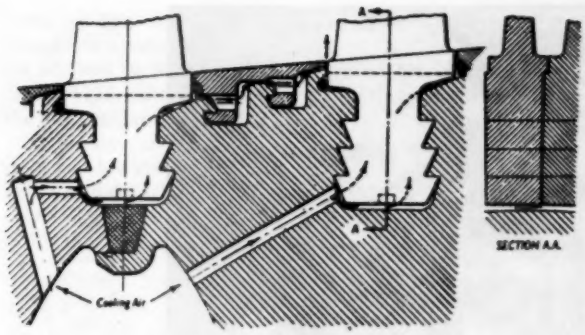

FIG. 16 DETAILS OF TURBINE BLADE-ROOT COOLING FITTED IN EL.60A

air flow, would enable ferritic rotors to be used in future designs of this type.

Conclusion

The EL.60A gas turbine did not achieve the original object of obtaining early sea experience with this new form of prime mover in a naval vessel. Nevertheless, many valuable lessons were learned, perhaps the most important being that heavyweight gas turbines constructed on steam-turbine lines are not suitable for naval machinery. In addition, a number of components were developed which could be used in future gas-turbine designs.

RM.60 MAIN PROPULSION GAS TURBINE (15,16)[7]

Introduction

The second development in this category originated when Messrs. Rolls Royce approached the Admiralty with the suggestion that a gas turbine would be a suitable propulsion engine for coastal craft. Investigations were carried out by Rolls Royce under the direction of the Engineer-in-Chief and a contract for the design and development of a 6000-hp Naval gas turbine (known as the RM.60) for use in coastal craft was placed in September, 1946. The life of the engine was stipulated as 1000 hr, comprising 300 hr at full power and 700 hr at 60 per cent power. It was later decided to install two of these engines in the gunboat *Grey Goose*. Design work started in December, 1947, and in June, 1951, the prototype RM.60 was on test. In the succeeding three months the engine completed 227 hr of trouble-free running during which period a power of 5300 hp was achieved.

Description of RM.60

In order to comply with naval requirements for economical low-power cruising, it was found necessary to employ a high pressure ratio, with intercooling between each major stage of compression and a heat exchanger. A photograph and a diagrammatic sectional arrangement of the complete engine are shown in Figs. 17 and 18, respectively. The engine comprises an 11-stage axial L-P compressor discharging through twin intercoolers arranged in parallel, each of which has a thermal ratio of 86 per cent. The air then passes to the H-P centrifugal compressor which is a two-stage design with an intercooler of 64 per cent thermal ratio between the stages. The H-P compressor discharges to a compact U-tube-type heat exchanger which incorporates a partial by-pass valve on the gas side. From the heat exchanger the air is led to twin combustion chambers and thence to the single-stage H-P turbine, the temperature of the gases being 827 C (1521 F). The gases then pass through intermediate (power) and L-P turbines, the output drive from the former passing through the H-P turbine and compressors. From the L-P turbine the exhaust gases pass through the heat exchanger or by-pass valve to the atmosphere.

The straight compound cycle with intercooling, heat exchange, and concentric turbines was chosen after an exhaustive analysis of various cycles and engine layouts. It is worth stressing that the comparison of different gas-turbine designs, though a most interesting occupation, is also a most dangerous one. Quite a small error in the assumptions, particularly of losses in various parts of the system, can be magnified in the ensuing argument and lead finally to totally wrong conclusions. Although reheat appeared theoretically very attractive in these analyses for decreasing the bulk and air rate, it was not employed in the RM.60 for a number of practical reasons, the most important of which are as follows:

(*a*) The difficulties of control, especially of maneuvering, would be increased to a serious degree since two separate fuel supplies must be regulated independently.

(*b*) Design of the reheat combustion chamber is difficult since it must work over a very large range of air/fuel ratios and the only cooling air available is already very hot.

(*c*) Reheat increases the temperature of the comparatively large L-P turbine where the stresses are already high.

Aero-engine practice has been adopted for the general principles of many components. This policy has resulted in a light and compact power unit with a specific weight (based on maximum power obtained on a test) of 5.3 lb/hp (including double-reduction gearing and all accessories).

Preliminary Development

In the development of a complex engine of this type, employing a higher pressure than any other open-cycle gas turbine, and with a gas temperature which at the time had been employed only in simple cycles, it will be appreciated that there was a considerable number of unknowns. This gave rise to extensive work on test rigs before the design of the various components could be finalized. One important requirement was for a high-pressure, high-temperature seal, and the type finally adopted, which has been used extensively throughout the engine, is shown in Fig. 19. The "concentric" design eliminated nearly all the problems with hot ducts (at the expense of a concentric shaft and buried bearings) but one that remained was to provide sufficient flexibility in the duct between the heat exchanger and the combustion chambers. This was solved satisfactorily by the design of flexible joint shown in Fig. 20, this type of joint being also used for the cold ducts connecting the H-P compressor, intercooler, and heat exchanger.

Another difficult problem was to design a burner which would cover the whole load range of the engine, necessitating a turn-down ratio of 30 to 1. This range is far too great for a simple pressure-jet burner and an ingenious triple burner was developed in which three concentric pressure-jet atomizers are brought into use in sequence as the pressure in the fuel line rises. The RM.60 was only required to burn distillate fuel, and the fuel system was therefore largely based on the use of aero-engine components.

The division of the engine into separate units lent itself to testing the H-P unit on its own, particularly as the power turbine was fitted immediately adjacent to the H-P turbine and was, in fact, part of the same unit. The H-P or "cruising unit" was first run in 1950 and 640-hr testing was carried out before assembling the complete engine. No major mechanical trouble was experienced but minor modifications were necessary to the blading of the power and compressor turbines to obtain correct matching and reduce proneness to surging.

Testing the Complete Engine

Development running of the complete RM.60 engine began in June, 1951, and on the first occasion of opening up the throttle a

[7] Appendix A, column 4; Appendix B, diagram 4.

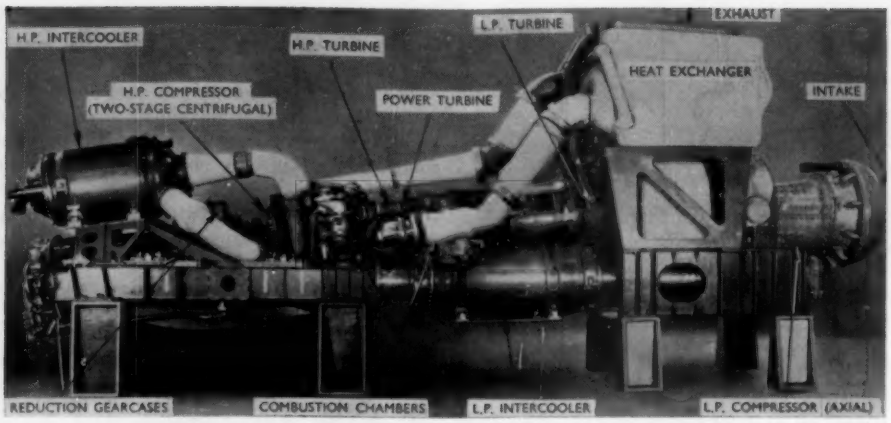

FIG. 17 RM.60 MAIN PROPULSION GAS TURBINE

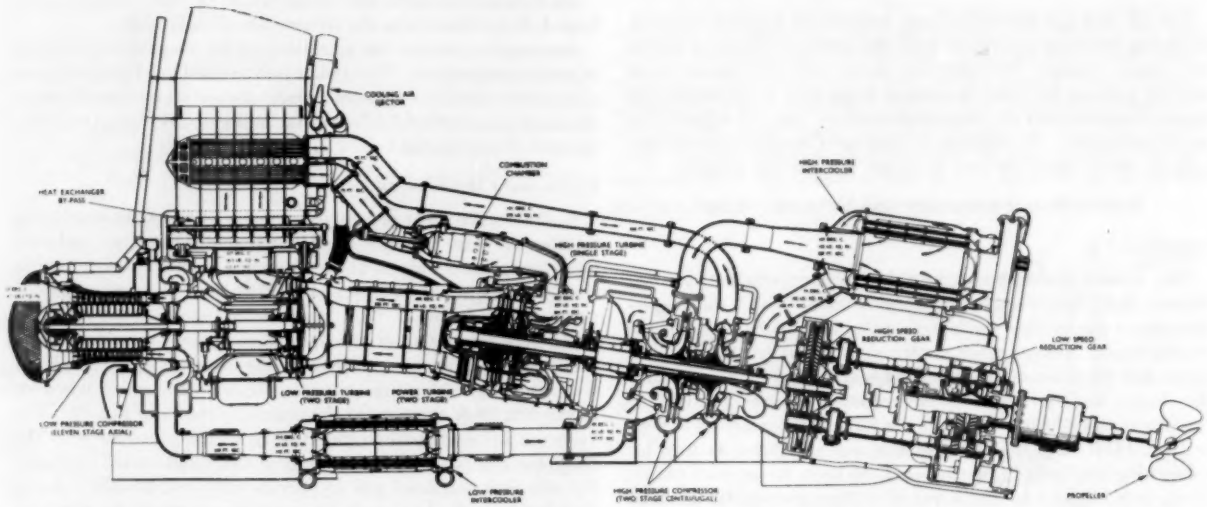

FIG. 18 DIAGRAMMATIC SECTION THROUGH RM.60 WITH DETAILS OF PRESSURES, TEMPERATURES, AND VELOCITIES THROUGH THE CYCLE

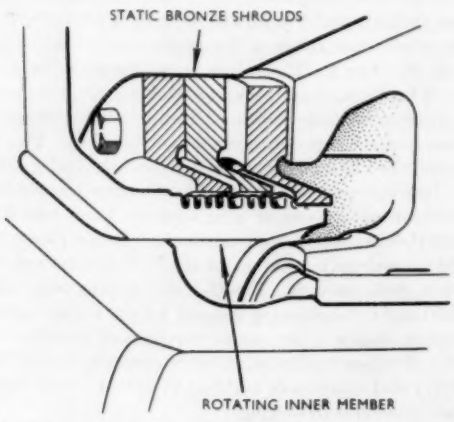

FIG. 19 DETAILS OF HIGH-TEMPERATURE, HIGH-PRESSURE SEALS USED IN RM.60

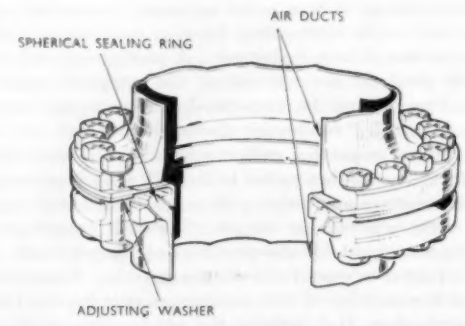

FIG. 20 DETAILS OF FLEXIBLE JOINT FITTED IN AIR DUCTING OF RM.60

power of 5300 hp was achieved. Testing of the engine continued over a period of two years and a total of 1100 operating hours (in addition to the 640 hr on the cruising unit) were amassed, during which time the characteristics of the engine were studied, the performance improved, and the reliability tested.

At the time of writing, two RM.60 gas turbines are being installed in the gunboat *Grey Goose* by Messrs. Vospers of Portsmouth and sea trials are expected to commence shortly. Fig. 21 shows a half-sectional model of the installation, and a line drawing giving the positioning of the engine in the hull is shown in Fig. 22.

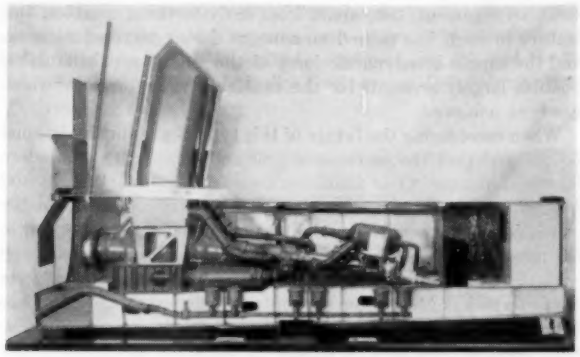

FIG. 21 MODEL OF RM.60 IN "GREY GOOSE"

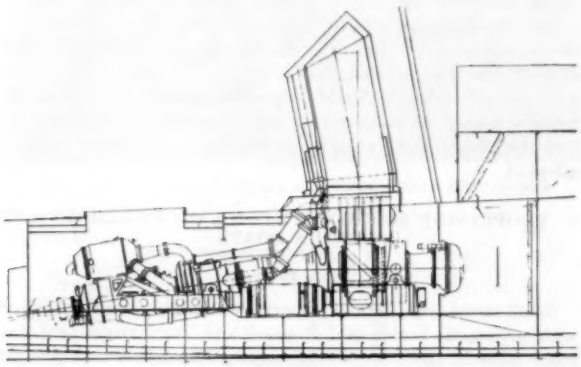

FIG. 22 DIAGRAM SHOWING INSTALLATION OF RM.60 IN "GREY GOOSE"

(In the actual boat installation the control room has now been relocated on the upper deck). By installing these engines in *Grey Goose* a considerable increase in power has been achieved with a reduction of 50 per cent in total machinery weight and a saving in space compared with the original steam installation, which was the lightest ever produced for naval purposes (17).

Performance Achieved

The specific fuel consumption achieved by the RM.60 engine during shore trials is shown in Fig. 2, the power with a 15 C (59 F) ambient air temperature being 5400 hp and with a 27 C (81 F) air temperature 4900 hp, these powers being obtained with the heat-exchanger by-pass open. This performance fell short of the original design estimate but, taking into account the advanced nature and complexity of the engine, the trial results have been most satisfactory. It has been established beyond doubt that the cycle chosen does enable a single gas turbine to maintain a good performance over a wide range of power. The rise of specific fuel consumption at the higher powers (see Fig. 2) emphasizes the price which must be paid if a small and compact heat exchanger incorporating a by-pass is used. For naval applications this price is acceptable since only a small percentage of a warship's operating life is spent at high power.

Experience Gained With RM.60

(*a*) *Maneuverability.* The engine is started by an electric motor rated at 40 hp for 30 sec. The whole starting sequence is automatically set in motion by a push button in the control room and the engine reaches idling speed in less than 30 sec. During shore testing of the RM.60 the rate of acceleration and deceleration has had to be limited to avoid overfueling or compressor surge. The time required to increase from idling to full power is about 45 sec, and about 40 sec is necessary to reduce from full power to idling. These times were obtained on the test bed with a "clean" engine, and the dashpot fitted in the throttle control for the ship installation has a 60-sec rate for both acceleration and deceleration, to allow some margin for fall off in compressor performance. This limitation on the flexibility of the RM.60 is mainly due to the low-inertia H-P turbine and compressor being more responsive to throttle movements than the high-inertia L-P unit. In future designs the handling could be considerably improved by incorporating larger blowoff valves at maximum cycle pressure, arranged to open when the throttle is suddenly closed. Nevertheless, the experience with RM.60 has shown the need for consideration in the early design stages of the naval requirements for rapid maneuvering.

(*b*) *Bearings.* The rotors of the three turbines and two compressors are carried in ball-and-roller bearings and during test-bed running these bearings have probably caused more concern than any other component. A number of failures have occurred and the H-P turbine thrust bearing has been particularly troublesome. It is extremely difficult to predict the value of the thrust on this bearing accurately and when surging occurs even its direction is uncertain. In the final design the capacity of the thrust bearing has been increased and the rate of control of the engine limited to avoid surging. Failures with other bearings have been cured by modifications to the cages and improved lubricating arrangements. Nevertheless, experience with RM.60 has again shown that ball-and-roller bearings cannot give the life and reliability required for the main propulsion gas turbines of naval vessels.

(*c*) *Erosion of H-P Compressor Impellers.* After the first 220-hr running, severe erosion of the aluminum impellers of the H-P centrifugal compressor was experienced. This was caused by the water droplets condensed from the atmospheric air in the intercoolers. Various anodizing and lacquering processes were tried without success and, as it was impracticable to fit a water separator, the impellers and diffuser vanes were replaced in stainless steel. This prevented erosion for the remainder of the test-bed running, but in salt-laden atmospheres further difficulties may arise and there undoubtedly will be material problems in the H-P compressors of intercooled naval gas turbines. It may be desirable to fit some form of water separator if a compact unit with low pressure loss can be developed.

(*d*) *Effect of Ambient Temperature.* Like all gas turbines the RM.60 is very sensitive to variations in ambient-air temperature and, in this particular case, sea-water temperature also, as it is an intercooled design. Fig. 23 shows this effect and it will be seen that power falls off rapidly above an ambient temperature of 27 C (81 F). (In the RM.60 the power is limited to 5400 hp below 15 C (59 F).) This large variation in power with ambient temperature raises problems in naval installations since warships normally must be capable of operating satisfactorily in arctic or tropical climates.

(*e*) *Heat Exchanger.* During one period of 320-hr running the thermal ratio of the heat exchanger deteriorated appreciably, due

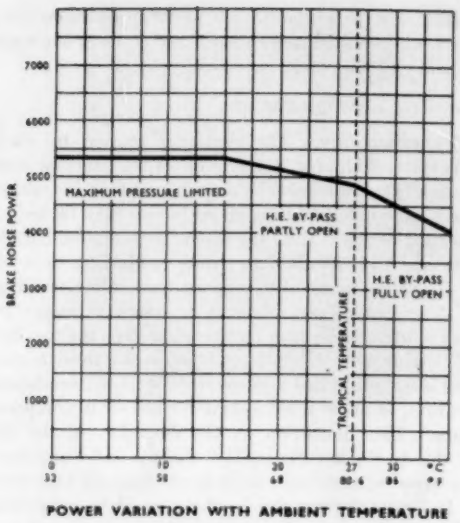

FIG. 23 RM.60. POWER VARIATION WITH AMBIENT TEMPERATURE

to the build-up of soft carbon on the gas side. This occurred during early-development running when combustion was by no means perfect. It is a pointer, however, that even when burning distillate fuel it may be necessary to clean the gas side of the heat exchanger at fairly frequent intervals.

(*f*) *Controllable Pitch Propeller.* Reversing of the two RM.60 gas turbines installed in *Grey Goose* will be carried out by means of Rotol 3-bladed controllable pitch propellers. Shore trials of the RM.60 have shown that very little improvement in performance can be obtained by operating at power turbine speeds away from the propeller law. Thus the original proposal to vary the pitch throughout the power range has been dropped and the propellers will be used for reversing only. It has also been decided that the propellers will reverse through the feathered position rather than through zero pitch. A mechanical stop will be fitted to prevent the blades taking any finer pitch than that required for ahead operation and, on reversing, the coarsening of the pitch to feather will act as a brake on the shaft system. Thus there will be no tendency for the output turbines or shaft system to overspeed when maneuvering.

(*g*) *Noise.* The RM.60, as built and tested, is undoubtedly noisy. The highest noise level of 120 decibels occurs at the H-P compressor and appears to be excited by the centrifugal impeller vanes and resonances from the small-bore ducts taking air at high velocity from the compressor. These ducts are very thin, and considerable noise reduction probably could be achieved by increasing the wall thickness and fitting insulation. When fitted with a closed intake duct the forward part of the engine is distinctly quieter at 110 decibels. It is considered that, with the experience from RM.60, a new design of engine should be possible giving a much lower all-round noise level without sacrificing performance.

Conclusions

At the time of writing the RM.60 has not been tried at sea so it would be premature to draw any sweeping conclusions on the future of this type of gas turbine for warship propulsion. Nevertheless, the first development project of this kind has produced an engine of 5400 hp with a specific weight of 5.3 lb/shp (including reduction gearing) and a fuel consumption below 0.65 lb/shphr over the whole range from 25 per cent to full power. These characteristics are suitable for a number of naval vessels but at present the life of the RM.60 limits its application to coastal craft. It is worth noting, however, that on the basis of normal warship operation the life of the hot parts would be nearer 10,000 hr than 1000 hr for the same maximum gas temperature.

To improve the life and reliability, plain journal and thrust bearings will be necessary causing increased weight and mechanical losses. The specific fuel consumption could be restored by the use of a larger heat exchanger, however, but the additional power absorbed by the bearings would be lost.

Criticism may be leveled at the complexity of the concentric-shaft arrangement, but, apart from early teething troubles, this feature in itself has caused no concern during test-bed running, and the simple aerodynamic form of the engine which it makes possible largely accounts for the satisfactory performance which has been achieved.

When considering the future of this type of gas turbine it must be realized that the increase in size and complexity of modern fighting equipment has made the need to reduce the weight and bulk of machinery increasingly important, particularly in smaller warships where machinery weight represents a large proportion of the total displacement. The range of powers normally associated with these smaller warships, say, 5000 hp to 15,000 hp, is precisely that in which neither steam turbine nor diesel machinery shows to best advantage. The lightweight diesel has not yet been successfully developed to meet these powers, while the specific weight of steam-turbine installations increases rapidly below 15,000 hp. It is in this field that the gas turbine is first likely to supplant existing prime movers as the sole propulsion machinery for warships.

The naval requirement for high efficiency over a wide range of power can be met either by a single complex gas turbine of the RM.60 type or by an installation comprising a number of simple gas turbines, each of which is designed to operate over its peak efficiency range. More operating experience will be necessary before deciding which of these arrangements ultimately will be adopted.

2 PROPULSION MACHINERY FOR LANDING CRAFT AND SHIP'S BOATS

INTRODUCTION

Small naval craft normally operate at a high percentage power with occasional bursts of full speed and, from the weight and space aspects, there are several applications where a simple gas turbine should show to advantage compared with existing diesel installations. (Owing to the fire hazard gasoline engines are not normally fitted in British naval craft.) The most promising of these applications are for short-range landing craft and ship's boats.

Until small gas turbines are in production and several have been tested afloat it is difficult to make a valid comparison between the two prime movers but the lightweight high-speed diesels at present in service require a considerable planned maintenance effort to insure their reliability, and the inherent simplicity of the small gas turbine should prove an advantage in this respect.

THE ROVER T.8 GAS TURBINE[8]

With a view to gaining operating experience with small gas turbines the Admiralty placed an order with the Rover Motor Company for four T.8 engines in July, 1950. Rovers were associated with Air Commodore Sir Frank Whittle's jet-engine development and the T.8 gas turbine was an early postwar venture designed for vehicle or boat propulsion. Although one engine was demonstrated by the company in a gas-turbine car during September, 1950, and two more in a launch the following month,

[8] Appendix A, column 5; Appendix B, diagram 5.

it was emphasized that the units were still entirely experimental and the four engines were sold to the Admiralty on this clear understanding.

The T.8 gas turbine is a simple-cycle unit with an independent power turbine and, in its original form developed 130 hp with a gas temperature of 800 C (1472 F). The heat exchanger provided for in the original design was not fitted since troubles were still being experienced with this component and the resulting gap was bridged by tubes. An integral gearbox reduced the power turbine speed in the ratio 7:1. A photograph and sectional arrangement of the T.8 gas turbine are shown in Figs. 24 and 25, respectively.

The four gas turbines were delivered early in 1951 after acceptance tests at the maker's works; one engine was installed in a 52½ ft harbor launch (No. 3964), shown in Fig. 26, and the remainder went to the Admiralty Engineering Laboratory, West Drayton, for shore testing.

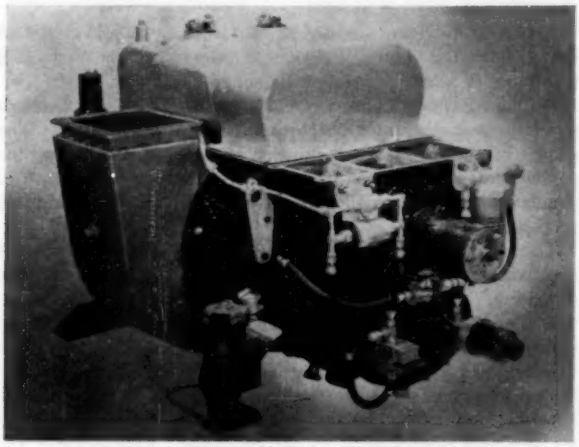

FIG. 24 T.8 GAS TURBINE FOR SMALL-BOAT PROPULSION

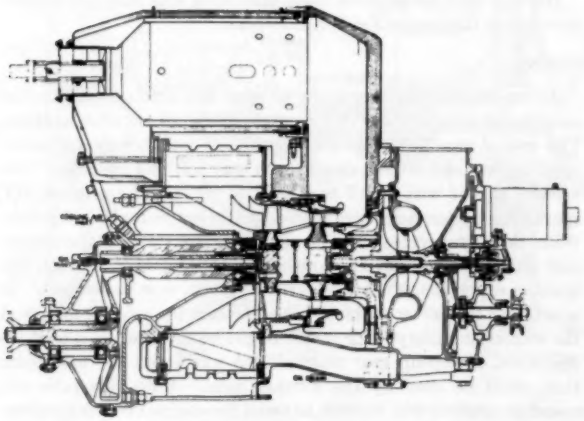

FIG. 25 SECTIONAL ARRANGEMENT OF T.8 GAS TURBINE

Harbor Launch 3964 (7)

Details of the T.8 gas-turbine installation in the harbor launch are shown in Fig. 27. The engine was mounted on "Silent-bloc" antivibration mountings and drove through a cardan shaft into a reversing gearbox which further reduced the shaft speed to 820 rpm at 100 hp. This gearbox was of a standard type suitably modified to eliminate the unlocked neutral position, thereby reducing the risk of overspeeding the power turbine. The air supply to the gas turbine was drawn through twin deck cowls (one of which is clearly visible in Fig. 26), containing perforated zinc splitters packed with fiberglass and passed down a trunk similarly lined. Exhaust gases from the twin ports passed into a common

FIG. 26 GAS-TURBINE-PROPELLED HARBOR LAUNCH NO. 3964
(Courtesy, Keystone Press Agency, Ltd.)

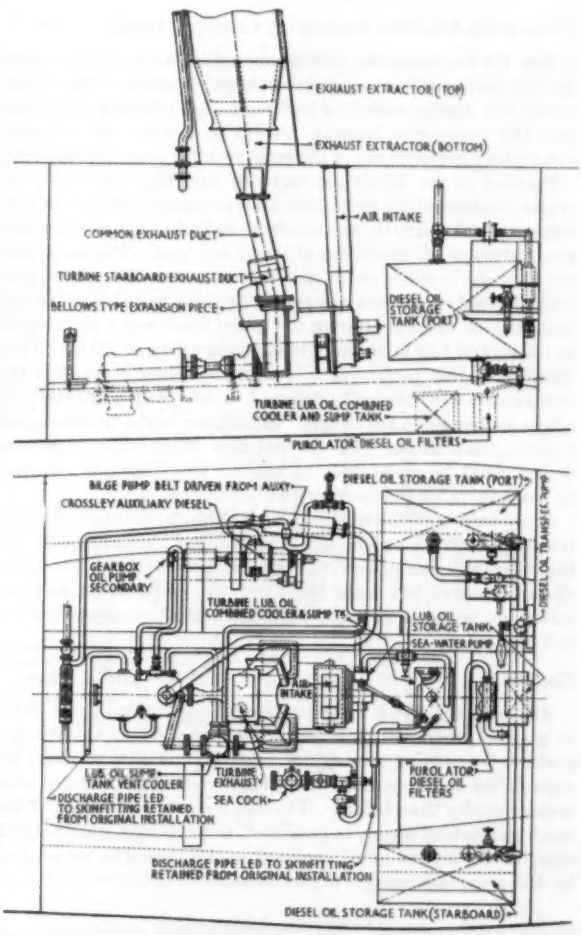
EXHAUST EXTRACTOR (TOP)
EXHAUST EXTRACTOR (BOTTOM)
AIR INTAKE
COMMON EXHAUST DUCT
TURBINE STARBOARD EXHAUST DUCT
BELLOWS TYPE EXPANSION PIECE
DIESEL OIL STORAGE TANK (PORT)
TURBINE LUB. OIL COMBINED COOLER AND SUMP TANK
"PUROLATOR" DIESEL OIL FILTERS
BILGE PUMP BELT DRIVEN FROM AUXY.
CROSSLEY AUXILIARY DIESEL
DIESEL OIL STORAGE TANK (PORT)
GEARBOX OIL PUMP SECONDARY
TURBINE LUB. OIL COMBINED COOLER & SUMP TK.
LUB. OIL STORAGE TANK
SEA-WATER PUMP
AIR INTAKE
LUB. OIL SUMP TANK VENT COOLER
DISCHARGE PIPE LED TO SKINFITTING RETAINED FROM ORIGINAL INSTALLATION
TURBINE EXHAUST
SEA COCK
"PUROLATOR" DIESEL OIL FILTERS
DISCHARGE PIPE LED TO SKINFITTING RETAINED FROM ORIGINAL INSTALLATION
DIESEL OIL STORAGE TANK (STARBOARD)

FIG. 27 DIAGRAM SHOWING INSTALLATION OF T.8 GAS TURBINE IN HARBOR LAUNCH 3964

trunk of aluminized mild steel and thence to the funnel via an exhaust extractor designed to ventilate the engine room. The Rover gas-turbine installation weighed 600 lb and occupied one third of the engine-room space required by the $2^1/_2$-ton diesel machinery originally fitted.

Trials of the harbor launch began in June, 1951, and no difficulty was experienced with the operation or control of the craft. Apart from an impeller-vane rub caused by wear in the main compressor-bearing housing, the mechanical performance of the T.8 gas turbine was satisfactory. The lubricating-oil consumption, however, rose steadily until after some 90-hr running it reached nearly 3 gph. By this time the shore testing of the remaining engines at the Admiralty Engineering Laboratory had brought to light other points requiring attention, and it was therefore decided to remove the gas turbine from the launch and return it, with the three others, to the maker's works for modification.

A modified gas turbine has now been reinstalled in the harbor launch and trials are continuing.

Trials at the Admiralty Engineering Laboratory (AEL)

The T.8 engines at the AEL were run for a total of 212 hr when the lubricating-oil consumption became excessive. This defect, which was finally traced to inefficient seals allowing oil to leak into the compressor housing, tended to obscure the otherwise satisfactory performance of the main rotating parts and bearings.

Starting of the Rover gas turbines was fully automatic, the engine reaching idling speed 7 sec after pressing the starter switch; response to the throttle was excellent and vibration was virtually nonexistent at all conditions of speed and load. The units were not excessively noisy and it was difficult to hear the turbine at all when a diesel engine was running in the same test shop! The fuel consumption obtained during the initial trials was 1.43 lb/shphr at 130 hp and 1.59 lb/shphr at the cruising rating of 100 hp. When considering this performance it must be remembered that the components were originally designed to deliver up to 200 hp with a heat exchanger in the circuit. In addition to the lubricating-oil trouble, trials at the AEL showed that other improvements in design could be made and the engines were therefore returned to the makers for the necessary modifications.

Shore testing of the modified engines has now restarted and it is evident that the new design of seals fitted has reduced the lubricating-oil consumption to negligible proportions. The other changes carried out have improved the performance and the maximum power is now 200 hp with a fuel consumption of 1.1 lb/hphr.

Conclusion

Although the T.8 is an experimental gas turbine never likely to go into production, the Navy has gained much valuable experience by running the engines, making it possible to assess the capabilities and appreciate the design of small gas turbines with greater clarity than before. The sole major disadvantage of the small gas turbine seems, as predicted, to be its high fuel consumption, but for several naval applications this should be outweighed by the many advantages it can offer.

Turbomeca Artouste[9]

During the trials of the Rover T8, publicity was given to a range of small gas turbines being developed by the Société Turbomeca at Bordes, France. The Artouste single-shaft engine and its corresponding free-power-turbine version, the Turmo, were of suitable power for landing-craft propulsion or emergency power generation and therefore of direct interest to the Admiralty. The extremely low weight of these units offered considerable advantages when compared with existing diesel installations. In addition, the Blackburn Aircraft Company had obtained a license to manufacture the Turbomeca gas turbines.

The Admiralty therefore placed an order in May, 1952, for two Artouste I engines for testing at the AEL. At this time the Turmo I was still under development but, since the two engines had many components in common, it was considered that experience with one would prove a reliable guide to the performance of the other.

Figs. 28 and 29 show a photograph and sectional arrangement, respectively, of the Artouste gas turbines as supplied to the Admiralty. The engines, which have a maximum rating of 276 hp, were of standard Turbomeca design and weighed 216 lb. The design consists of a centrifugal compressor, annular combustion chamber, and axial-flow turbine with a built-in gearbox reducing the output shaft speed to 6000 rpm.

A most interesting feature is the method of injecting fuel into the annular combustion chamber through radial holes in a rotating disk forming a part of the hollow turbine shaft.

On the AEL test bed the Artouste gas turbine drives through a reduction gearbox and clutch to a water dynamometer. The sluices of the latter are motor-driven and operated from the engine control position so that, if desired, testing conditions can simulate those of a small craft fitted with a controllable pitch propeller.

Testing started on November 13, 1952, and an aggregate of approximately 750 running hours has been achieved at the time of writing.

Preliminary results indicate that the Artouste was capable of stable operation over the whole power/speed range likely to be required for generator or propulsion applications (with a controllable pitch propeller). The engine was designed to burn aviation kerosene but the combustion with gas oil was smoke-free and stable at all powers with no carbon deposit whatsoever. The maximum continuous rating was put at 250 shp and at this rating the fuel consumption was 1.04 lb/shphr when burning gas oil.

Having established these facts, attention was directed toward developing the engine for Naval Service.

Starting

As supplied, it was necessary to start the Artouste on gasoline and change over to main fuel approximately 10 sec after ignition. The use of two fuels was not considered satisfactory for naval applications and direct starting on gas oil was specified. To achieve this it was found necessary to replace the original HT spark plug by torch igniters with a small electric fuel boost pump. With this arrangement the normal time for starting the engine and putting it on load is about 20 sec. Concurrently with the ignition experiments a hand-starting system was developed. It was found that after making due allowance for cold oil drag and the various auxiliary loads which might be expected in a boat installation, a step-up gear ratio of 78.5 was about the maximum that could be used by the average man. With this ratio the speed of rotation was too fast to assist the engine over its acceleration "flat spot" from 5000 to 7000 rpm. A three-speed bicycle gear was therefore fitted and it is now possible for one man to hand-start the engine with comparative ease.

Control

The control system fitted to the Artouste was designed for generator applications. For boat propulsion (using a CP propeller) it was necessary to develop an additional control unit incorporating adequate safeguards to prevent damage to the gas turbines by unskilled operators.

[9] Appendix A, column 6; Appendix B, diagram 6.

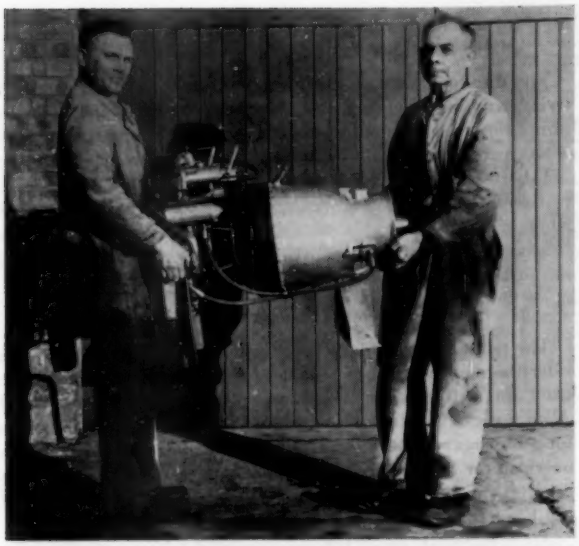

FIG. 28 276-HP ARTOUSTE GAS TURBINE AS SUPPLIED TO THE ADMIRALTY ENGINEERING LABORATORY

haust temperature is reached. Other safeguards required are two interlocks, one to prevent starting with the clutch engaged, and another to insure that the propeller remains in neutral pitch until the clutch has been engaged and the engine brought up to speed.

A simple and compact control box incorporating the foregoing features has been designed at the AEL and is now in use on the engine test bed. It consists of a single lever, used only on starting, which gives sequential operation of the fuel cock, clutch, and speed governor, and a handwheel for propeller pitch, used in subsequent maneuvering and operation.

Silencing

A disadvantage of the Artouste gas turbine is the noise level which is unacceptably high far most naval applications, and silencers are essential. Measurements taken at AEL with an absorption-type silencer on the exhaust showed a reduction in noise level from 111 db to 99 db at full power. Future work will be concentrated on silencing the compressor intake.

Running Experience

At the time of writing some 750-hr running has been carried out on the two Admirality Artouste I gas turbines. Apart from periodic injection of kerosene to wash the compressor, the routine

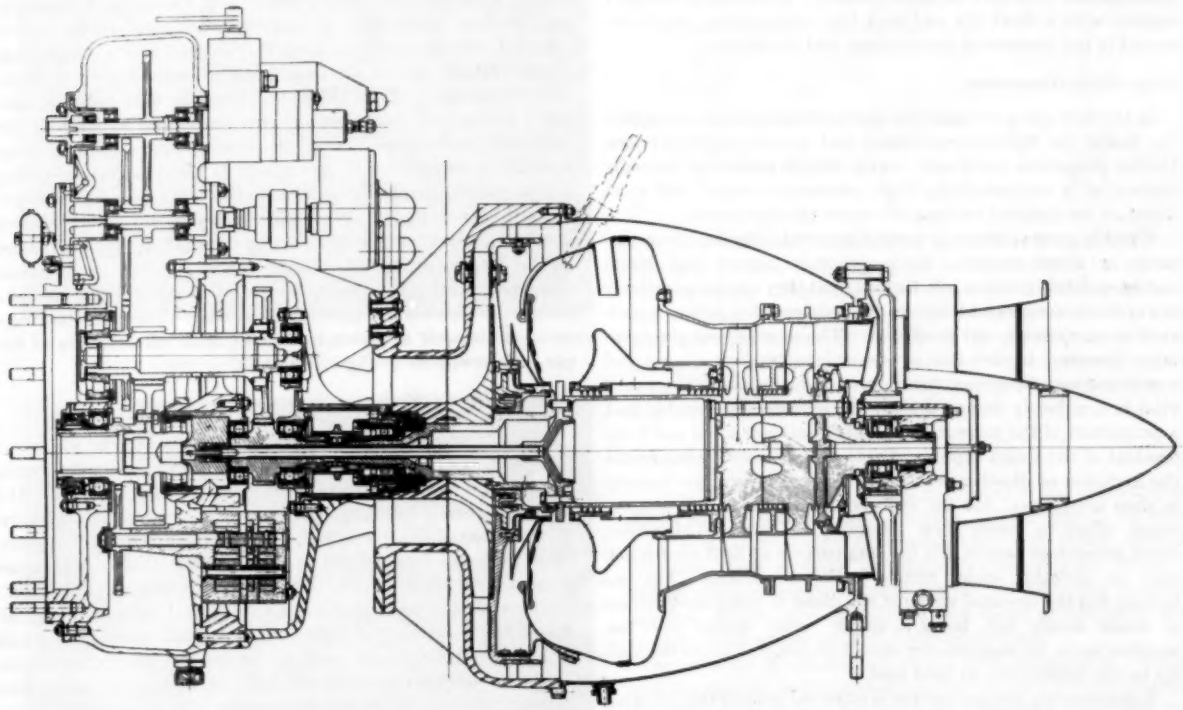

FIG. 29 SECTIONAL ARRANGEMENT OF ARTOUSTE GAS TURBINE

The rate of fuel input during acceleration is automatically controlled by a built-in device which also incorporates a lubricating-oil pressure trip and an overload trip operating at a predetermined fuel flow. This latter is not an adequate protection against overheating, since it does not allow for variations in air-inlet temperature or fouling of the compressor. For this reason, and to prevent complete shutdown by the overload trip which might endanger the craft, an additional control unit is necessary for reducing propeller pitch automatically when a limiting ex-

maintenance has been surprisingly small and the lubricating-oil consumption negligible.

Stripping of the engines has revealed minor design deficiencies which can be rectified easily, such as fretting of the turbine-bearing-support pins. The only serious defect has been scaling of the turbine blades and casings. This confirms the maker's fear that the scaling resistance of their turbine materials might not be adequate when burning gas oil (average sulphur content 0.8 per cent). No similar defect has occurred in Rover turbines

operating at higher temperatures with the same fuel and it is considered that the use of normal British materials would eliminate the trouble. The combustion chambers remained in perfect condition throughout and these components are certainly one of the most attractive features of the design.

Conclusions

The Turbomeca Artouste is an extremely light and compact gas turbine with a fuel consumption which is most creditable for its size. If British materials are used for the turbines, no difficulty is foreseen in adapting the engine to suit naval requirements for short-life propulsion engines and generators of the emergency type; but before the evaluation can be regarded as in any way complete many more running hours are required to establish life and reliability.

3 AUXILIARY MACHINERY

Introduction

There are a number of naval auxiliary applications where gas turbines have potential advantages over other prime movers. These can be divided broadly into continuous-running machines where low fuel consumption and comparatively long life are important, and emergency engines which run at very infrequent intervals and then only for short periods. In this latter category engines with a short life and high fuel consumption can be accepted in the interests of compactness and simplicity.

Main Electric Generators

In the first category come the main electric generators supplying power for fighting equipment and general ship's services. Unlike propulsion machinery, naval electric generators normally operate at a comparatively high percentage output and must therefore be designed for long life under this condition.

Warship generators are at present powered either by steam turbines or diesel engines. It is generally agreed that steam turbogenerators perform satisfactorily but they are comparatively heavy in themselves and require the services of a boiler, associated steam systems, and auxiliaries. The most serious disadvantage, however, is that loss of steam involves loss of electrical power and war experience showed that in many cases this proved vital in a seriously damaged ship. It is therefore desirable that a proportion of the generators should be self-contained and independent of the steam supply. Up to the present this has meant the inclusion of diesel generators, which are heavy, particularly in sizes above, say, 350 kw and require a considerable maintenance effort to insure their reliability in service. Moreover, diesel generators cannot idle for long periods at light load which may be desirable under certain action conditions. The gas turbine has the essential virtue of the diesel in being independent of steam supply but, being a simple rotary engine with few working parts, its maintenance should be less, and it has the ability to idle indefinitely on light load.

To summarize, the gas turbine is expected to have the following advantages over existing prime movers for the main generating machinery of warships.

1 Compared with steam turbogenerators:

(*a*) Reduced weight and space (for whole installation).

(*b*) Independence of steam supply.

(*c*) Quicker starting.

2 Compared with diesel generators:

(*a*) Reduced weight and space (particularly in the larger size).

(*b*) Reduced maintenance.

(*c*) Ability to run on light load for long periods.

(*d*) Reduced lubricating-oil consumption.

The disadvantages of gas turbogenerators are that comparatively large inlet and exhaust ducts are required, the fuel consumption is nearly double that of a diesel (though comparable with steam machines), and distillate fuel is at present necessary.

In a steam-driven warship, therefore, it would appear desirable to have a combination of steam and gas-turbo main generators, while in a diesel or gas-turbine-driven warship, gas turbogenerators may have advantages over diesel generators, particularly in the larger sizes (above, say, 350 kw).

Allen 1000-Kw Gas Turboalternator (1, 2, 18, 19)[18]

The potential advantages outlined led the Admiralty to consider the development of a naval gas turboalternator and a contract for a 1000-kw machine was placed with Messrs. W. H. Allen Sons & Co. Ltd., in April, 1948. The specification called for a maximum continuous rating of 1000 kw with a 20 per cent overload for 10 min, allowing for a loss of 0.6 psi in the air-inlet duct, 0.4 psi in the exhaust duct, and under temperate 15 C (59 F) or tropical 38 C (100 F) conditions.

After carefully balancing the conflicting requirement of low weight and space with reasonably good efficiency, a comparatively simple open-cycle set was designed having an axial compressor of $4^1/_4$ to 1 pressure ratio driven by a two-stage turbine, an annular heat exchanger, a multichamber combustion system disposed symmetrically around the engine, and a single-stage power turbine driving the alternator through Allen-Stoeckicht epicyclic gearing. Plain sleeve-type bearings were used throughout. A separate power turbine was introduced to improve the part-load performance in the normal service operating range from 50 to 80 per cent of power, and to prevent the set stalling due to sudden overloads under action conditions. This complicates the governing problem, however, and to assist the speed governor during sudden load changes a small flywheel is fitted on the output turbine shaft (immediately adjacent to the reduction gear) while the compressor rotating assembly is specially designed to have a low moment of inertia. A sectional arrangement of the unit is shown in Fig. 30, while photographs of the gas turbine appear in Figs. 31 and 32.

Trials at Firm's Works

Trials (without heat exchanger) started in March, 1951, and full speed, full load, and overload were achieved in the first series of tests with the minimum of mechanical teething troubles. The work was greatly facilitated by a consultancy agreement with the Bristol Aeroplane Company, Ltd., which entitled Messrs. Allen to design data, advise on production methods, and the use of certain testing facilities. Testing was continued at the firm's works with the heat exchanger for a total period of 860 hr. Apart from mechanical defects in the heat exchanger, which will be discussed, there were no major troubles. The results of the preliminary governing trials are of interest since they demonstrate that a gas turboalternator with free power turbine and heat exchanger is capable of governing within close limits.

Detailed results are given in Table 1.

In addition, the set will accept instantaneous load changes of 75 per cent without the emergency gear operating and modifications are in hand to increase this to 100 per cent.

During the latter part of the trials at Messrs. Allen there was a steady fall-off in performance which could not be accounted for other than by leakage. On opening up the heat exchanger a ring securing the heat-exchanger spacer to the power-turbine casing was found to be sheared and as a consequence the tube

[18] Appendix A, column 7; Appendix B, diagram 7.

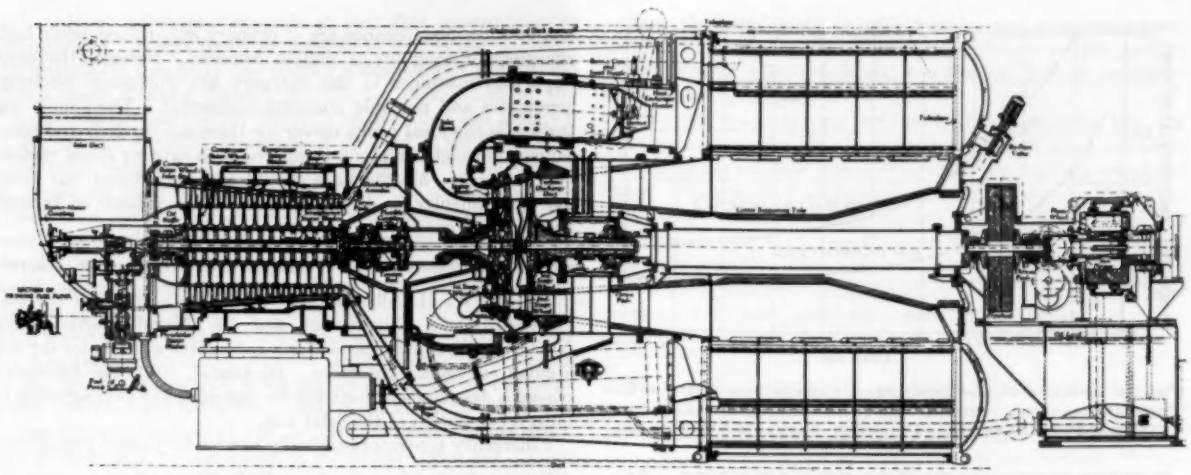

FIG. 30 SECTIONAL ARRANGEMENT OF 1000-KW GAS TURBINE

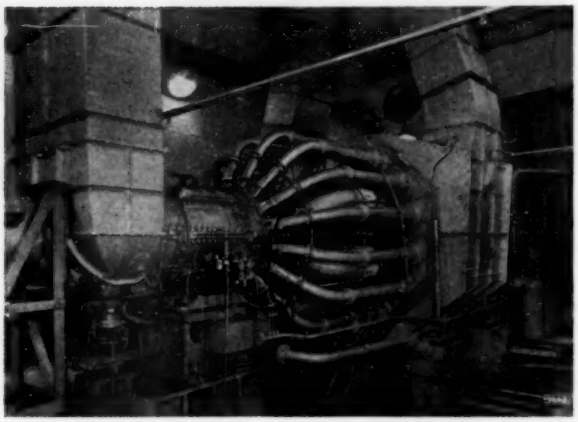

FIG. 31 1000-KW GAS TURBOALTERNATOR IN TEST HOUSE. VIEW FROM COMPRESSOR END WITH CASINGS REMOVED

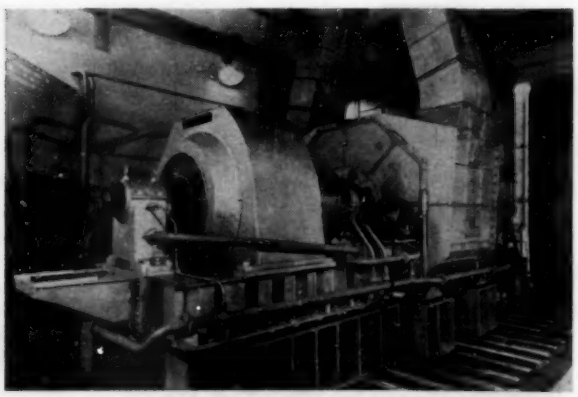

FIG. 32 1000-KW GAS TURBOALTERNATOR IN TEST HOUSE. VIEW FROM ALTERNATOR END

plate was retained only at its outer diameter. The air pressure caused the unsupported tube plate to buckle and the innermost two rows of tubes released their grip and the ferrules projected through the plate as illustrated in Fig. 34. This, of course, accounted for the leakage and loss of performance. Another

TABLE 1 RESULTS OF SHOP TESTS ON ALLEN GAS TURBO-ALTERNATOR

Condition	Measured fluctuation in speed, per cent
Gradual load changes (per cent) between	
0 and 100	1.2 max
Instantaneous increase of load (per cent) from	
0 to 25	—4.6
25 to 50	—2.5
50 to 75	—2.4
75 to 100	—2.2
Instantaneous decrease of load (per cent) from	
100 to 75	+1.55
75 to 50	+2.1
50 to 25	+2.3
25 to 0	+2.9

defect revealed on completion of the 860-hr testing was heavy scaling of the heat-exchanger tubes in the vicinity of the incoming exhaust gases. The material of the tubes is aluminum bronze and the nominal steady temperature of the gases at entry to the heat exchanger about 480 C (896 F) though this may have risen momentarily as high as 650 C (1202 F). The mean wall thickness of the tubes is 0.018 in. and the loss of material due to scaling was in the region of 0.002 in.

Trials at Admiralty Test House

The 1000-kw set has now been re-erected at the Admiralty test house in the National Gas Turbine Establishment, Pyestock, with a stronger design of heat-exchanger retaining ring and defective tubes replaced in alternative copper-base alloys. A full program of testing is being carried out including endurance running, further governing trials, and minor development work to make the machine suitable for shipboard use.

At the time of writing, 817 hours have been run at Pyestock and the specific fuel-consumption curves obtained are shown in Fig. 33. (The curve for the heat exchanger build has been corrected to allow for leakage which has again been experienced.)

Testing at the Admiralty Test House has revealed the following points:

(*a*) The heat exchanger requires modification to give greater freedom for differential expansion. A better method of securing the tubes in the end plates and fewer casing joints also appear desirable.

(*b*) Without expensive instrumentation it is difficult to establish whether a heat exchanger is leaking. This difficulty will be very real in actual service.

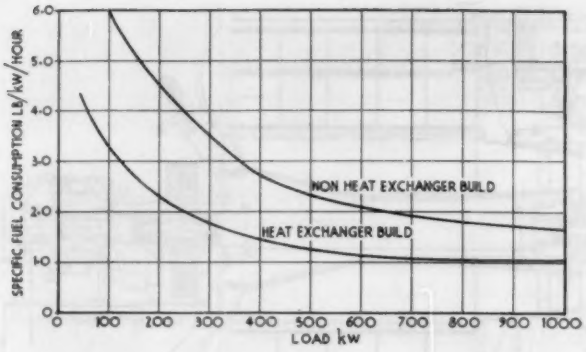

FIG. 33 SPECIFIC FUEL-CONSUMPTION CURVES FOR 1000-KW GAS TURBOALTERNATOR. ACTUAL TEST RESULTS USING ADMIRALTY DIESEL FUEL (GAS OIL), CALORIFIC VALUE 18,500 BTU/LB

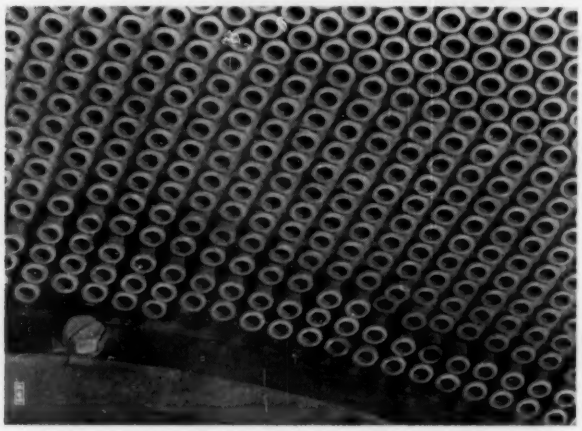

FIG. 34 DETAILS OF HEAT-EXCHANGER TUBE DEFECTS IN 1000-KW GAS TURBOALTERNATOR

(*c*) Initially, there were occasional failures of the combustion chambers to "crosslight" when starting. Larger interconnectors have been fitted and lighting up with the present system, with ignition provided by low-tension glow plugs, has been very good.

(*d*) Some auxiliaries are of the aircraft type and the life of these components does not appear adequate for main generator applications.

(*e*) The general mechanical design (heat exchanger excepted) and the combustion system have proved extremely successful. When action has been taken to rectify (*a*) and (*d*) the set should be capable of operating for very long periods without maintenance.

Conclusions

Initial experience with this first naval gas-turboalternator design has been most encouraging. There is every indication that a thoroughly reliable and comparatively efficient machine can be developed for an all-up weight (gas turbine only) of 10.5 lb/hp. Further trials at the Admiralty test house will be necessary but it is also desirable to obtain early operating experience at sea. An order has therefore been placed with Messrs. W. H. Allen for two additional gas turbogenerators, each of 500 kw capacity, which it is intended to install afloat in the near future.

EMERGENCY ENGINES

There are numerous naval requirements for emergency or intermittent equipment where lightweight, compactness, quick starting, and low maintenance are of primary importance, while high efficiency or long life is seldom required. Perhaps the most important machines in this category are emergency electrical generators and portable pumping equipment. The simple gas turbine is the ideal prime mover for these applications and great savings in weight and bulk compared with existing diesel engines can be made. Maintenance should also be reduced but there may be difficulty in disposing of the large volume of exhaust gases in some cases.

ALLEN EMERGENCY GAS TURBINE FOR GENERATOR DRIVE[11]

In 1950 W. H. Allen & Sons, in collaboration with the Engineer-in-Chief's Department and the Naval Marine Wing NGTE, began a design study for a gas turbine suitable for driving a small emergency generator. In August, 1951, the Admiralty placed a development contract for one unit, the firm agreeing to meet a proportion of the total cost.

Emergency generators normally run for very short periods and

[11] Appendix A, column 8; Appendix B, diagram 8.

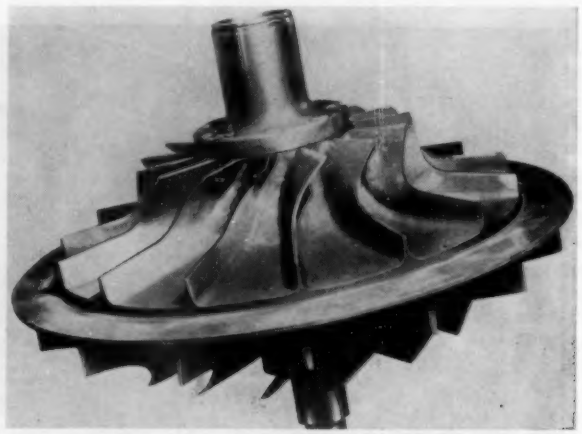

FIG. 35 SMALL EMERGENCY GAS TURBINE. COMBINED COMPRESSOR AND TURBINE ASSEMBLY

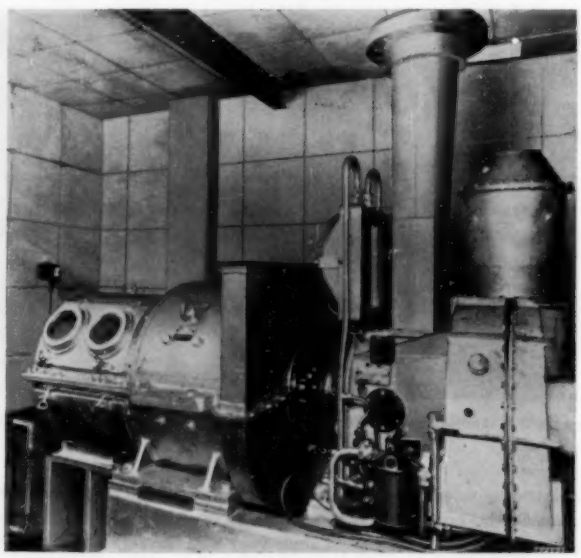

FIG. 36 SMALL EMERGENCY GAS TURBINE AND GENERATOR IN TEST HOUSE

then only at infrequent intervals so that fuel consumption is unimportant. It was therefore possible to reduce the design to the ultimate in simplicity and employ a simple-cycle gas turbine with a single rotating element, Fig. 35. A photograph of the Allen emergency gas turbine driving a generator on test is shown in Fig. 36 and a section-arrangement drawing of the unit (with combustion chamber omitted) appears in Fig. 37. In the interest of robustness plain bearings are fitted and the centrifugal compressor and inward radial-flow turbine are machined from the same forging of Rex 448 steel. Originally, the complete rotor was overhung and was designed to run above the first critical speed under normal conditions. During the initial trials at the end of 1952, however, it was found that the amplitude of whirl was so great as the critical speed was approached that the rotor rubbed on the casing. It was therefore decided to change to the present design with a bearing on either side of the rotor which now runs well below the critical speed under all conditions.

Development testing of the engine continued during 1953, and at the time of writing a total of 763 hr have been run. From the mechanical point of view the engine has performed very satisfactorily and no major defects have been experienced, but the power output and thermal efficiency are below the design estimates (see Appendix A, column 8). The maximum power obtained at the designed gas temperature of 730 C (1346 F) during the initial tests was 123 hp with a specific fuel consumption of about 2.0 lb/hphr. The main reason for the disappointing performance was considered to be a poor design of compressor-inlet casing resulting in an over-all compressor efficiency some 10 per cent below the estimated value.

After the initial trials a modified casing and compressor rotor were fitted which have raised the power output to 180 hp but the compressor efficiency is still too low and further modifications are being considered.

For naval emergency gas turbines it is most desirable that the starting arrangements should be entirely self-contained and independent of any external sources of supply. Preliminary tests have therefore been carried out recently, during the development running, with a Rotax cartridge starter. This unit consists of two 720-gram cordite cartridges firing on to a small turbine wheel. The tests have been very successful, the cartridge starter taking only 3 sec to accelerate the gas turbine to 900 rpm when it pulls away under its own power up to synchronous speed.

Conclusion

Further development will be necessary before the designed performance is achieved, but an extremely simple, robust, and durable emergency gas turbine has been produced which should prove valuable for many naval emergency applications. It appears that the inward radial-flow turbine fitted in this set is on the upper limit for size and for larger units an axial turbine would be preferred.

Gas-Turbine Portable Pumps

Another emergency application where a lightweight gas turbine has considerable advantages is for portable salvage or fire pumps. To investigate this application the Admiralty have very recently placed an order with the Rover Motor Company for three "Neptune" gas-turbine-driven portable pumps. A

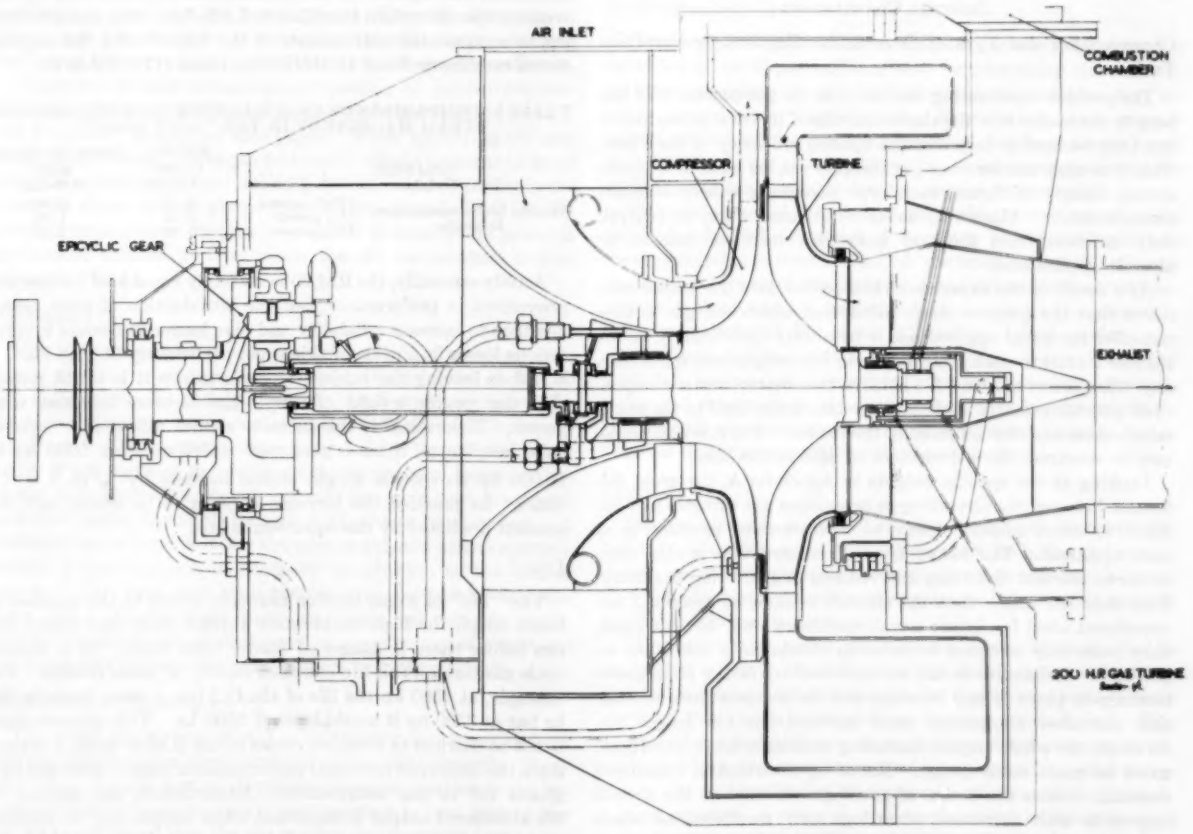

Fig. 37 Sectional Arrangement of Small Emergency Gas Turbine Without Combustion Chamber

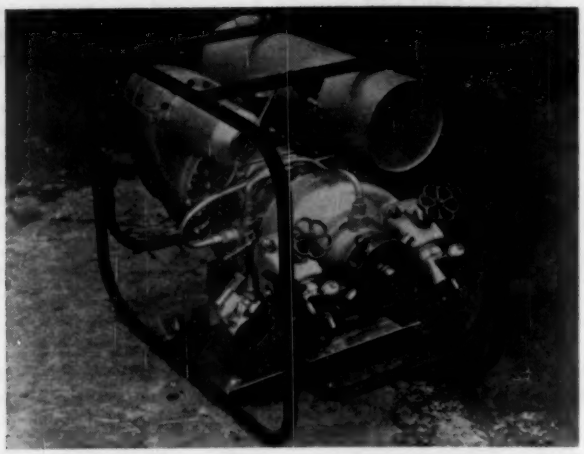

FIG. 38 NEPTUNE GAS TURBINE-DRIVEN PORTABLE PUMP

photograph of one of these pumps (without hand-starting equipment) is shown in Fig. 38. The output of the unit is 500 gpm at a delivery pressure of 100 psi when pumping from a static lift of 10 ft. The dry weight complete with frame, fuel tank, hand-starting mechanism, and fully instrumented panel is approximately 200 lb. The pump will normally be hand-started and the suction line is primed by an air ejector fed from the gas-turbine compressor.

GENERAL CONCLUSIONS

Characteristics and Type of Construction Required for Naval Gas Turbines

The problem confronting the Navy in the gas-turbine field has been to determine how the characteristics of this new prime mover can best be used to improve the fighting efficiency of the Fleet. This is no easy matter since gas turbines can be built in a bewildering variety of forms, each type possessing widely different characteristics. Moreover, naval requirements are, in general, very different from those of industrial, merchant-marine, or aircraft applications.

As a result of the experience obtained to date the author considers that the greatest single advantage which the gas turbine can offer for naval applications is its ability (particularly in the simpler forms) to pack more power in less weight and space than any other prime mover. To achieve this characteristic an open-cycle plant is essential and this type also lends itself to the maximum amount of development in the future. But it is also necessary to construct the gas turbines on lightweight lines.

Looking at the specific weights in Appendix A, the great difference between the aircraft-type propulsion gas turbines and the EL.60A, which adheres closely to steam-turbine practice, is at once apparent. The heavy type of construction is also more prone to thermal distortion and reduces maneuvering flexibility. This does not imply that the aircraft designs, as described, are considered ideal for future naval machinery and, in most cases, they were only accepted in order to obtain early operating experience. Reliability would be improved by fitting plain sleeve bearings in place of ball bearings and the temperatures, stresses, and corrosion allowances need modification for longer life. In short, the whole engine (including auxiliaries and control gear) must be made more robust. But it is, nevertheless, considered essential to base the design of naval gas turbines on the aircraft approach with increased scantlings and modifications where necessary than to follow in the tradition of established steam practice. Unless this is done the important military advantages of reduced weight and space, rapid starting, and great flexibility in operation may never be realized (20).

Assuming that the design of future naval gas turbines is based on the lines indicated, the major factors likely to influence their use in warship applications will be discussed.

Performance

The high fuel consumption of the simple gas turbine, particularly at low powers, is clearly shown in the performance curves of the Gatric and G.2 in Fig. 2. These engines are based on aircraft designs which are now over 10 years old, however, and with modern components, fuel consumptions below 0.7 lb/shphr at full power could be obtained with reasonable life. To effect this improvement a higher compression ratio is necessary which tends to reduce flexibility and may create compressor-blade vibration problems. By splitting the compressor (twin-spool design) higher compression ratios can be achieved with improved flexibility and fuel consumptions below 0.6 lb/shphr could now be obtained with this type of design. These developments will not alter the steeply rising performance characteristics of the simple gas turbine, however, and consumption figures at half load will remain some 25 per cent higher than at full power. Simple engines on their own are therefore intrinsically unsuitable for applications where economy at low power is required.

To obtain higher efficiencies and, more particularly, to improve part-load performance a more complex cycle with heat exchange and intercooling is necessary. Referring again to Fig. 2, the great superiority in part-load efficiency of the RM.60 compared with the G.2 should be noted. This improvement has been obtained at the expense of considerable complication and an increase in specific weight from 2.28 to 5.3 lb/hp. It is also interesting to compare the performance of the RM.60 with the original steam machinery fitted to HMS *Grey Goose* (17) (Table 2).

TABLE 2 COMPARISON OF RM.60 GAS TURBINE WITH ORIGINAL STEAM MACHINERY IN THE "GREY GOOSE"

		RM.60	Steam machinery
Total SHP		10,800	8000
Weight		5.3 lb/hp	14 lb/hp
Specific fuel consumption, lb/shphr	Full power	0.66	0.88
	1/2 power	0.61	1.03
	1/4 power	0.76	1.56

Aerodynamically, the RM.60 is certainly capable of further improvement in performance, but the introduction of plain sleeve bearings to increase reliability and give longer life would involve serious losses in a cycle of this type, particularly at part load.

Before leaving the subject of performance it is worth noting that the specific weight of simple gas turbines increases with power. Theoretically, this increase should follow the "square-cube law;" e.g., if an engine were scaled up from 5000 hp to 10,000 hp its specific weight should increase by $\sqrt{10/5} = 1.4$ times. In practice the increase appears to be about half the amount predicted by the square-cube law.

Life

The "life" of a gas turbine normally refers to the number of hours which the high-temperature turbine disks and blades will run before there is danger of failure from creep. In a simple-cycle gas turbine this life increases rapidly at lower powers. For example, at 4500 hp the life of the G.2 (on a creep basis) is 300 hr but at 3500 hp it would be over 2500 hr. This characteristic is not so marked in complex cycles of the RM.60 type, however, since the improved part-load performance is mainly achieved by a slower fall in gas temperature. Nevertheless, the increase of life at reduced output is important when considering the application of gas turbines to naval machinery. It is doubtful, for example, if many warships exceed 2000 hr at full power during

their whole life and in some cases this figure may be as low as 500 hr. For naval propulsion machinery, therefore, the full-power life can be extremely short judged by industrial standards and there is only a requirement for longer life at reduced output. Even main generators seldom operate at full power and, although detailed records are not available, the majority of running probably occurs in the range between 50 and 80 per cent power.

In the present state of gas-turbine development combustion-chamber liners, bearings, seals, auxiliaries, etc., usually fail before the engines are "life-expired" on a creep basis. From the mechanical point of view, however, there is no fundamental reason why the lightweight type of construction advocated in this paper should not be capable of giving reliability and long life in all components except the "hot parts." The overhaul periods would then be controlled by the combustion-chamber liners and the "creep life" referred to.

Effect of Ambient Temperature

It is well known that gas-turbine performance is extremely sensitive to ambient conditions and stated figures of power or efficiency mean little unless associated with a definite temperature and pressure. Fig. 23 shows the power variation in the RM.60 with changes in air and sea temperature. In the case of the G.2 the maximum power reduces from 4500 to 4000 hp with an increase in inlet-air temperature from 15 C (59 F) to 27 C (81 F). This large variation in performance may give rise to difficulties in naval vessels which must operate satisfactorily in arctic or tropical climates. The size of gas turbine will usually be fixed by the output specified for tropical conditions, and in cold climates considerably more power will theoretically be available at the same gas temperature. To take advantage of this power it is necessary to stress the turbine, gearing, shafting, and propellers accordingly, which results in a heavier installation.

There are obvious advantages in quoting all gas-turbine performance figures at a standard temperature and a figure of 15 C (59 F) is used throughout this paper. When specifying the ambient conditions for naval gas turbines it should be borne in mind that propulsion machinery can only be run near full power in the open sea where the air temperature seldom exceeds 33 C (91 F), whereas main generators may be required to develop full power in landlocked tropical harbors where the air temperature is considerably higher.

Duct Sizes (1)

Gas turbines require much larger quantities of air than other prime movers. Compare, for example, the air rate of RM.60 (70 lb/hphr) or the G.2 (87 lb/hphr) with average figures for diesel machinery or steam-turbine plant (11 and 14 lb/hphr, respectively). This situation is aggravated by the greater sensitivity of gas turbines to inlet and exhaust pressure losses. (The power falls approximately 10 per cent for each 1 psi inlet pressure loss and by 5 per cent for each 1 psi exhaust pressure loss. A pressure loss of 1 psi in either the inlet or exhaust causes approximately 5 per cent loss in efficiency in addition to the loss of power.) The large duct sizes necessary in gas-turbine installations are a disadvantage since they take up valuable space, and the increased size of deck opening are a source of structural weakness and increase the vulnerability to battle damage.

Reliability and Maintenance

The naval gas turbine is still in its infancy and many defects and teething troubles have occurred in the early designs. Basically, it is the simplest of all engines, however, and when the accumulated experience to date is incorporated in new designs the author is convinced that the gas turbine (in its simple forms) can be made superior to all other prime movers as far as reliability and maintenance are concerned, without sacrificing its very high power-weight ratio. Moreover, the small size of components will greatly facilitate their removal and replacement for maintenance purposes. The more complex cycles will inherently have somewhat less reliability but should compare favorably with other engines. It also appears that a heat exchanger designed for minimum weight and space may add appreciably to the maintenance commitments.

Flexibility and Control

The majority of gas turbines described in this paper have independent (free) output turbines, giving flexibility of torque and speed over the whole power range. This characteristic is essential for propulsion with fixed-pitch propellers, but at idling speed some residual torque will always be available, equivalent to about 3 per cent of full output. For generator applications a free-power turbine improves part-load efficiency, and enables the machine to accept sudden overloads in emergency without stalling; but the governing problem is greatly increased.

A single-shaft gas turbine delivers no power below about 30 or 40 per cent of maximum speed and this type will normally be confined to constant-speed machines such as generators. It does appear feasible, however, to operate a single-shaft design as a propulsion engine with a controllable pitch propeller but this arrangement would be most uneconomical for low-speed cruising.

It has been effectively demonstrated in the Gatric, G.2, and Rover T.8 installations that gas-turbine controls can be reduced to a throttle lever and starting switch. This simplicity has obvious operational advantages.

Production Cost (1)

An assessment of production cost at this early stage in the evolution of naval gas turbines may be misleading since manufacturing experience is very limited and any new design may involve considerable development. Nevertheless, the present cost of manufacturing a proved design of simple engine in small numbers should not exceed £15 per hp, which compares favorably with other prime movers ordered to special requirements.

The production of lightweight gas turbines involves precision machining to high standards of accuracy but is well suited to mass production in time of war. If the numbers justify full-scale tooling-up the cost of simple-cycle naval gas turbines should approach that of aircraft propeller turbines which are currently being manufactured for about £7 per hp. High-performance gas turbines contain certain quantities of so-called "strategic" materials, but nowadays nearly all materials have some strategic value and it appears to the author that, on a horsepower basis, the over-all strategic content for gas turbines is little higher than for much heavier conventional machinery.

Fuel

The gas turbines described in this paper will operate satisfactorily on a wide range of distillate fuels, including Admiralty diesel (gas oil) and aviation kerosene, but they can only burn residual fuel for short periods in emergency and then only if suitable heating equipment is provided.

There are many naval applications where distillate fuel is acceptable but it is generally assumed that before gas turbines can replace steam installations for the main propulsion of large warships they must be capable of burning residual fuel. The burning of residual fuel in open-cycle gas turbines presents formidable problems, the most important being deposition and corrosion of the turbine blades by inorganic constituents in the ash. A large program of research and development is being directed to the solution of these problems and it appears that by controlling combustion and using small quantities of additives in the fuel,

both deposition and corrosion ultimately may be reduced to acceptable limits. At the time of writing, however, little practical experience in burning residual fuels at the high temperatures necessary for naval gas turbines exists.

Reversing (2)

In steam installations reversing is provided by a separate astern turbine which idles in the condenser vacuum during ahead running. There is no such vacuum in a gas turbine so that windage losses would be excessive. In addition, there would certainly be difficulty in keeping the control valves gastight at high temperatures and the complicated ducting necessary might compromise ahead performance. If gas turbines are to be used as the sole means of propulsion some external means of reversing is therefore essential. Possible methods of providing this reversing are as follows:

(*a*) Electric drive.
(*b*) Controllable pitch propellers.
(*c*) Mechanical reversing gears.
(*d*) Hydraulic reversing couplings.
(*e*) Separate lightweight gas turbine for astern use only.

A detailed discussion of these methods is outside the scope of this paper but it is worth mentioning that (*e*) is only practical when the astern power required is a small fraction of the ahead power. Experience with (*b*) will be obtained in *Grey Goose*, and (*c*) has been used successfully in harbor launch No. 3964.

Noise

High-intensity noise appears to be inherent in the design of gas turbines approaching the required standards of lightness for naval applications. The majority of the noise emanates from the compressor inlet and the exhaust, where silencing can be readily applied. At the compressor inlet the noise is usually of high frequency consisting of fundamental and harmonics of the number of blades per row times the speed of rotation. Noise from this source can be reduced simply and effectively by fitting splitters and by lining the inside of the duct with sound-insulating material. In MGB.2009, for example, this procedure resulted in a noise reduction of 39 decibles from the original level of 117 decibels. At the exhaust, noise covers a wide band of frequencies and can best be silenced by reducing gas velocities, by fitting torpedo-type splitters, and by lining the funnel with silencing material.

Future of Gas Turbine in Royal Navy

The gas turbine will surplant existing machinery in warship applications only where it is shown to be generally superior based on the special naval requirements listed earlier in this paper. The only true criterion for comparison is service experience at sea, and this is, unfortunately, still extremely limited, but sufficient knowledge has been gained from the projects described to indicate the future of the gas turbine in the Royal Navy with some confidence (21).

In the author's opinion, gas turbines will be introduced in increasing numbers for the propulsion of high-speed craft. In major warships the first applications will be as "boost" units for use at high powers with steam turbines or possibly diesel machinery for cruising. At a later stage gas turbines may become the sole means of propulsion in some warships. Gas turbines also will be found in certain landing craft and ships' boats in the near future.

Turning next to the auxiliary-machinery field, gas-turbine generators for normal or emergency use will be fitted in increasing numbers and gas-turbine portable pumps introduced for fire fighting and salvage.

There are other naval applications not yet fully explored, where the characteristics of the gas turbine may show to advantage. Among these are boiler blowers and the provision of low-pressure air for salvage and other purposes.

Looking further into the future, it must be remembered that the first tentative steps in British naval gas-turbine development were taken just over 10 years ago, and this new prime mover is therefore still only on the threshold of development. Component efficiencies will certainly increase in the future and, when cooling techniques allow higher gas temperatures to be used, it is the author's opinion that gas turbines will find very wide applications in naval vessels. The consequent saving in weight and space, the reduced maintenance, and the rapid starting and maneuvering will greatly increase the fighting efficiency of the Fleet. In fact, the partial supplanting of existing machinery by gas turbines may well prove as revolutionary as the change from reciprocating engines to steam turbines at the turn of the century.

Acknowledgments

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This paper is published with the approval of the Lords Commissioners of the Admiralty but the responsibility for any statements of fact or opinion expressed herein rests entirely with the author.

BIBLIOGRAPHY

1 "The Marine Gas Turbine Plant in 1951," by Comm. W. T. Sawyer, USN, Trans. SNAME, vol. 59, 1951, pp. 454–487.

2 "Gas-Turbine Progress Report—Naval Vessels," by Capt. W. A. Dolan, Jr., USN, and Lieut. A. S. Hafer, USNR, Trans. ASME, vol. 75, 1953, pp. 185–199.

3 "Gas Turbine Propelled M.G.B. 2009," *The Engineer*, Part I, September 5, 1947, pp. 218–220; Part II, September 11, 1947, pp. 248–249; and Part III, September 19, 1947, pp. 261–262.

4 "A Naval Pioneer," *Oil Engine and Gas Turbine*, September, 1947, pp. 170–172.

5 "Installation of Gas Turbine," by A. T. G. Coleborn, *Marine News*, vol. 35, October, 1948, pp. 36–46.

6 "Gas Turbine Propelled M.G.B. 2009," by A. T. G. Coleborn, *The Engineer*, June 25, 1948, pp. 621–624.

7 "Naval Gas Turbine Developments," *The Engineer*, June 29, 1951, p. 861.

8 "The Combustion of Residual Fuels in Marine Gas Turbines," *The Engineer*, April 18, 1950, p. 511.

9 "Naval Gas Turbines for Intermittent Use," *Oil Engine and Gas Turbine*, February, 1953, p. 378.

10 "G.2 Naval Gas Turbine," *The Engineer*, January 9, 1953, pp. 58 and 59.

11 "The Prospects of Gas Turbines in Naval Applications," by Capt. R. T. Simpson, USN, and Comm. W. T. Sawyer, USN, *Mechanical Engineering*, vol. 72, 1950.

12 "Combination Propulsion Plants for Naval Vessels," by Comm. J. J. McMullen, USN, presented at the Semi-Annual Meeting, ASME, Los Angeles, Calif., July, 1953.

13 "A British 6,500 S.H.P. Long Life Marine Unit," *Oil Engine and Gas Turbine*, March, 1953, p. 398.

14 "A 6,500 H.P. Naval Gas Turbine," *The Engineer*, February 13, 1953, pp. 238 and 239.

15 "Rolls-Royce 5,400 S.H.P. Naval Gas Turbine," *Oil Engine and Gas Turbine*, November, 1953, pp. 279–282.

16 "R.M. 60. Marine Gas Turbine," *The Engineer*, November 13, 1953, pp. 617–621.

17 "Steam Gunboat Machinery—A Light Weight Steam Plant," by Comm. (E) H.A.K. Lay, RN, and Comm. (E) L. Baker, D.S.C., RN. Paper read before a joint meeting of the Institution of Naval Architects and the Institute of Marine Engineers, September, 1948.

18 "A Gas Turbine Alternator Set for the Royal Navy," *Oil Engine and Gas Turbine*, November, 1951, p. 281.

19 "1000 K.W. Gas Turbine Alternator Set for the Royal Navy," *Engineering*, November 16, 1951, pp. 609–613.

20 "The Marine Gas Turbine From the Viewpoint of an Aeronautical Engineer," by A. H. Fletcher, Proceedings of The Institution of Mechanical Engineers, series A, vol. 166, 1952, no. 2.

21 "The Gas Turbine as a Prime Mover on U. S. Naval Ships," by W. M. M. Fowden, Jr., R. R. Peterson, and J. W. Sawyer, presented at the Annual Meeting of the ASME, New York, N. Y., November 29–December 4, 1953.

22 "The Part-Load Performance of Various Gas Turbine Engine Schemes," by D. H. Mallinson and W. G. E. Lewis, Trans. The Institution of Mechanical Engineers, vol. 159 (War Emergency Issues 37–46), 1948, pp. 198–219.

Appendix A

DESIGN DETAILS OF BRITISH NAVAL GAS TURBINES

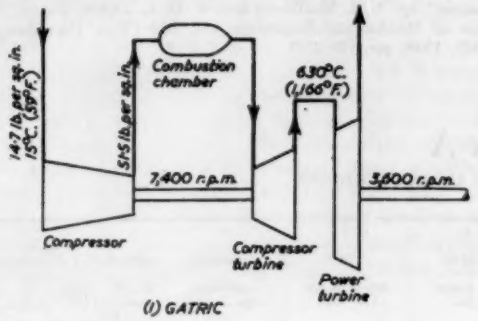
Column Number		1	2	3	4	5	6	7	8
Engine		Gatric	G2	E.L.60A	R.M.60	T8	Artouste I	1,000kW.G.T.A.	EmergencyG.T.A.
Maker		Metropolitan Vickers and Co., Ltd.	Metropolitan Vickers and Co., Ltd.	English Electric Co., Ltd.	Rolls Royce, Ltd.	Rover Motor Co., Ltd.	Turbomeca (Blackburns)	W. H. Allen, Sons and Co., Ltd.	W. H. Allen, Sons and Co., Ltd.
Maximum power at 15 deg. C. (60 deg. F.) air inlet temperature, h.p.		2,500	4,500	6,500	5,400	200	276	1,512	200
Duty		Boost propulsion	Boost propulsion	Main propulsion	Main propulsion	Craft propulsion	Utility	Main generator	Emergency generator
Cycle (see Note 1)		1/LP	1/LP	1/P/E	2SC/IP/IE	1/LP	1/C	1/LP/E	1/C
Turbine inlet temperature at maximum power, deg. C. deg. F.		750 1,382	800 1,472	704 1,299	827 1,521	847 1,557	800 1,472	650 1,202	750 1,382
Overall pressure ratio		3·5	4·0	4·02	18·5	4·0	3·7	4·25	2·65
Air flow at maximum power, lb. per sec.		47·5	65·6	128	64·6	3·5	4·75	37·1	5·7
Thermal efficiency at maximum power		12·8	17·2	20·4	20·4	12·5	13·75	19·75	12·9
Thermal efficiency at 50 per cent power (see Note 2)		9·5	13·4	17·75	22·6	9·2	9·8	15·6	7·5 (Design values)
Specific weight, lb. per h.p. at maximum power (see Note 3)		2·77	2·28	27·2 excluding bedplates	5·3	2·25	0·78	10·5	3·0
Designed life of hot parts, hr. (Based on creep data)		300	1,000	10,000	1,000	800	500 (estimated)	50,000	1,000
Compressors	Number and type	1 axial	1 axial	1 axial	1-l.p. axial 1 h.p. 2-stage centrifugal	1 centrifugal	1 centrifugal	1 axial	1 centrifugal
	Number of stages	9	11	15	11/2	1	1	13	1
Turbines	Number and type	2 axial	2 axial	2 axial	3 axial	2 axial	1 axial	2 axial	1 radial
	Number of stages	2/4	1/3	6/6	1/2/2	1/1	2	2/1	1
	R.P.M.÷1,000	7·4/3·6	7·83/5·2	4/5·6	15/11/7·18	40/35	35	8/6·75	23·5
Number of intercoolers and thermal ratio		None	None	None	3 (l.p. 2 in parallel): l.p. 86 per cent, h.p. 64 per cent	None	None	None	None
Heat exchanger thermal ratio, per cent		None	None	75	48 at full power with bypass open	None	None	70	None
Number and types of combustion chambers		1 annular	1 annular	6 cans	2 cans	1 Single can	1 annular	8 cans	1 Single can

NOTES:

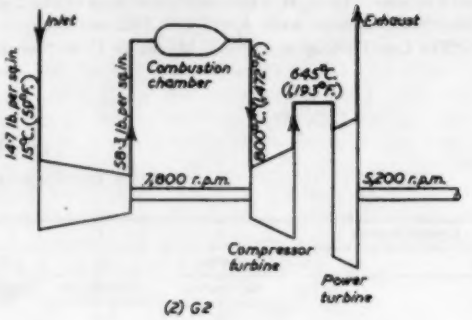
1 Cycle notation adopted from Mallinson and Lewis' paper, reference (22).
2 With the exception of column 8, thermal efficiences are actual values obtained on test using Admiralty diesel fuel (gas oil), calorific value 18,500 Btu/lb.
3 Specific weights quoted include reduction gearing (when fitted) and all engine-driven auxiliaries.

Appendix B

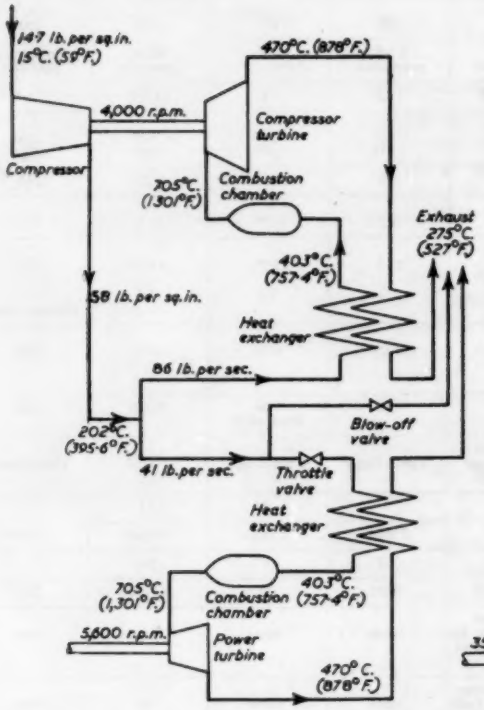
CYCLE DIAGRAMS FOR ADMIRALTY GAS TURBINES



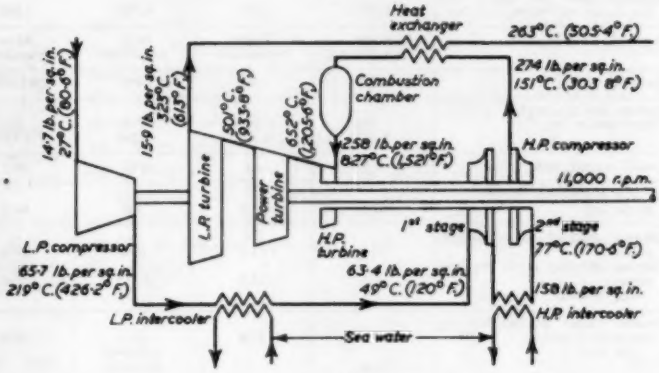
(1) GATRIC



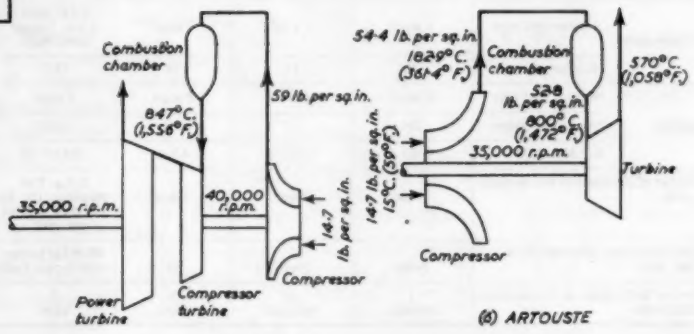
(2) G2



(3) EL 60A

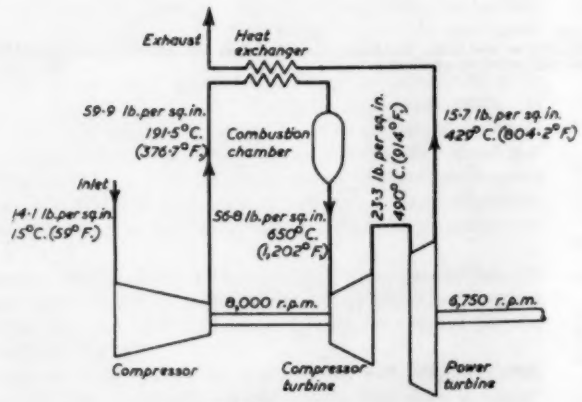


(4) R.M. 60

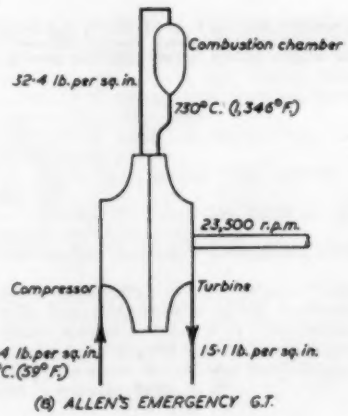


(5) ROVER T8

(6) ARTOUSTE



(7) 4,000 kW. G.T.A.



(8) ALLEN'S EMERGENCY G.T.

NOTE: Pressures and temperatures shown are design figures; pressures are absolute.

Discussion

W. M. M. Fowden, Jr.[12] The author has chronicled here for all gas-turbine enthusiasts and others, the major achievements and shortcomings of this new prime mover as applied to naval vessels. His paper will probably become a master reference for all those who proceed with the courage of their convictions in the development and application of marine gas-turbine engines.

In the lessons learned on each of the applications discussed by the author, it is interesting to note:

(*a*) The failure of antifriction bearings, especially after installation in the ship.

(*b*) The reduced maneuverability, i.e., acceleration and deceleration, associated with complex-cycle machines.

(*c*) The effect which hull vibrations can have on extremely lightweight propulsion machinery.

It is presumed that the required characteristics listed by the author under his General Conclusions are not in order of importance. In any case, it is desired to comment in some detail on those characteristics which have proved to be most important for U. S. Naval ships.

Reliability and Maintenance. The U. S. Navy has approximately 200 production gas-turbine engines installed in modern minesweepers. These engines have operated an estimated 40,000 hr since their installation. We have every indication that the reliability of these engines will be a maximum and the maintenance a minimum.

The writer concurs with the author that complex-cycle engines will be less reliable than simple-cycle machines. Since reliability is the most important characteristic of a naval prime mover, does he consider the complex-cycle gas turbine to have any future in this field?

Performance. In 10 years the performance of simple-cycle engines has improved tremendously. Specific fuel consumptions have been reduced as much as 50 per cent and existing simple-cycle engines commonly have an SFC of 0.8 lb/hphr. The author predicts efficiencies for modern engines in the vicinity of 0.6 lb/hphr. Would he care to predict or hazard an extreme estimate as to the efficiency of a simple-cycle engine operating for marine purposes in 1960?

Life. The characteristic entitled "life" is debated very learnedly throughout engineering intelligentsia with each participant using his own definition. The author appears to emphasize "creep" as the basis for engine life. The writer is sure that he also has considered "stress to rupture" as a basis for determining engine life. Engine life cannot be divorced from the application selected. Cyclic loading of extreme variation and frequency will reduce the life below that anticipated from the design. Experience will guide us in establishing this requirement; in the meantime, engines must be subjected to vigorous inspections at brief intervals lest we overlook the consequences of severe thermal shock.

Reversing. The U. S. Navy has successfully utilized mechanical reversing gears and controllable pitch propellers for reversing vessels with diesel propulsion engines. Special emphasis on the reversing methods (*b*) and (*d*) listed by the author must be made if the gas-turbine engine is to be applied successfully in naval ships.

The future of the gas turbine for naval ships as set forth by the author cannot be underestimated. It can rightfully be underlined and concluded with an "Amen."

[12] Lieutentant Commander, USN, Head, Gas Turbine Branch, Bureau of Ships, Navy Department, Washington, D. C.

A. A. Hafer.[13] This paper effectively demonstrates the tremendous progress which has been made in the naval gas-turbine field. It will undoubtedly be a valuable reference on naval gas-turbine operating experience for many years.

Sometimes we are inclined to overlook just how versatile a tool we have in the gas turbine. This is an engine that can do more than merely deliver shaft power. It has three other products available for utilization, namely, compressed air, which may be extracted from the engine compressor, exhaust heat which may be used for generating steam or heating feedwater, and highly preheated exhaust air which may be used for combustion air since it still contains about 85 per cent of the oxygen. When some or all of these products are utilized in addition to the shaft output, phenominally high over-all thermal efficiencies are attainable.

We have been prone in the gas-turbine field to hang back from making marine applications today, since something better is always just around the corner. When we ship a steam turbine out the back door we always have a superior model on the drawing board, but that does not prevent an attractive application of today's model. If we wait for the ultimate in a gas turbine we shall wait forever, because as a representative of a gas-turbine manufacturer, we expect to always bring out better and better machines as time goes on.

It is no accident that the aircraft gas turbine has become supreme in the military aviation field. Likewise, it will be no accident that the gas turbine will assume an important place in the naval field. The timing of this will be dependent only upon how clever naval engineers, naval architects, and gas-turbine builders are in exploiting the advantages which the gas turbine now offers.

In reference to the 200-hp Allen gas turbine the statement is made: "It appears that the inward radial-flow turbine fitted in this set is on the upper limit for size, and for larger units an axial turbine would be preferred." Does this refer to physical, mechanical, or aerodynamic considerations? Also with reference to the 1000-kw Allen machine, it is indicated that a flywheel was installed on the output-turbine shaft to facilitate governing. We have found and demonstrated by operating experience other effective governing means for two-shaft gas turbines which do not involve the weight and space sacrifices which the flywheel must. Does the author still feel it necessary or desirable to use a flywheel for a two-shaft electrical generation gas turbine? Another question is in reference to operating personnel. The writer understands the RM.60 is to be installed in a destroyer-escort type of vessel. This class of vessel had a normal watch for the steam plant of about 16 to 20 men under wartime conditions. How many operating personnel will a gas-turbine plant similar to the RM.60 require?

L. W. Shallenberg.[14] The British program is of particular interest to us who are in the field of gas-turbine development and application as it contains well-co-ordinated projects, each representing a great deal of advanced thought and planning.

The author is to be congratulated for an excellent paper delineating the British program. Further, both the author and his countrymen are to be commended for their rapid progress in gas-turbine development and application and in ferreting out and solving problems connected therewith.

By way of amplification, the writer would like to discuss certain results obtained on tests conducted at the Naval Engineering Experiment Station; in particular, results concerned with both

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[14] Commander, USN, United States Engineering Experiment Station, Annapolis, Md.

noise reduction and compartment ventilation. The station has performed a certain amount of work on minesweepers. Prior to the design and application of noise-reduction measures noise levels of 119 db were measured in the area of the compressor air inlet and 110 db in the area of the exhaust system. Application of silencing means similar to those discussed by the author permitted a reduction to 97 db at the compressor air-inlet system and 104 db near the exhaust. In terms of apparent loudness this amounts to about 42 per cent reduction.

On the Solar T-520 engine which is now being installed in the MSB's, we have made certain noise-level measurements. One of these engines is now on test at the Station. Noise levels of 120 db were measured 6 in. from the combustor outlet dropping to 96 db 3 ft from the engine and 91 db 6 ft from the engine. The Station's experience on these and other installations clearly bears out the author's conclusions that gas turbines definitely can be silenced and by relatively simple means.

Turning now to the ventilation of compartments. The Station conducted work in this field initially in connection with the MSB with the knowledge gained thereon being applied recently to other installations. The Station found that by means of an eductor-exhaust arrangement it was possible to pump some 3700 cfm of ventilating air in a stack handling 5500 cfm of combustion air (both at atmosphere conditions) or, in other words, a ratio of about 0.67 ventilating air to combustion air. It was found practicable to maintain average compartment temperatures about 20 deg F above that of ambient air. The ventilation blowers originally planned for were thereby eliminated.

It is noted that the author lists a number of ways of reversing a gas-turbine-propelled ship, one of the ways being hydraulic reversing couplings. Has any work been done on this particular kind of reversing system and if so, in what manner are the reversing couplings employed?

P. A. Pitt.[15] The author is to be complimented on the excellence with which he covered the numerous applications of gas turbines in the British Royal Navy. The experience of the Admiralty is an important contribution in helping to orient the gas turbine in its proper place relative to other types of prime movers.

The methods used to silence the Gatric were very effective. Does the author have information on the comparisons in noise level of the unsilenced turbine with the Packard engines in the MGB.2009? Would he care to comment on the Admiralty's viewpoint on the effect of noises from the gas turbine on the human being? And, whether specific noise limits have been established?

The Admiralty's experience with ball-and-roller bearings is very informative and will be a helpful guide in planning the gas turbine, particularly in large horsepower sizes. Does the author have an estimate of the expected power losses in modifying the RM.60 to sleeve bearings?

With regard to the Allen 1000-kw gas-turbine-alternator set it is interesting to note that part-load characteristics were placed before speed regulation in selecting two-shaft turbine design. Experience has shown that the single-shaft machine is far superior in holding speeds under severe changes in load as occurs in alternator sets. For example, the Solar T-400 and T-520 generator sets consistently hold speeds within 4.5 and 3 per cent, respectively, with 100 per cent electrical load changes. Their regulation could be controlled even more closely with further refinement of the governor.

In the single-shaft gas-turbine power plant, the turbine, plus the compressor, especially if the compressor is an axial-flow type, provide an exceptionally stiff rotational inertia system, which is very effective in absorbing transient loads and preventing stall. With the exception of large steam-turbine plants, the single-shaft gas turbine is probably superior to any other prime mover in holding alternator frequency. Does the author have any comment on the effect of thermal lag of the regenerator used in the Allen 1000-kw set on stabilizing speed during large load changes? What maximum sudden load changes does the Admiralty specify in setting up generator-set specifications?

G. L. Graves, Jr.[16] This paper is a realistic appraisal of the present status and future prospects of naval gas turbines.

In general, the writer feels that gas-turbine power-plant progress is largely dependent on the substantial improvements in performance which can be obtained by going to higher temperatures and pressures. Present-day gas turbines are limited by the stress limitations of current high-temperature materials. Even the best of these high-temperature materials doesn't have the strength at high temperatures that the more common materials have at lower temperatures. Improvements in the strength properties of high-temperature materials in the foreseeable future will be small, costly, and generally at the expense of other mechanical properties. Furthermore, these high-temperature materials contain large percentages of critical and strategic materials that are sometimes rather difficult to form.

More specifically, for the naval gas turbine, higher temperatures mean lower fuel consumption, fewer pounds of air required per horsepower (small ducts) and more horsepower per pound of engine weight. In order to achieve higher gas turbine-inlet temperatures, reduce quantities of critical and strategic materials, and permit the utilization of lower temperature, higher strength materials, the development of cooling methods for gas-turbine engines should be given a high priority within any gas-turbine development program. Furthermore, the utilization of lower temperature, higher-strength alloys in a cooled engine may even result in longer life for engine hot parts due to increased stress safety factors; however, this is something that will have to be evaluated in terms of blade design and fabrication methods.

The Bureau of Ships has had several contractors working on the development of compact heat exchangers for small gas turbines which were relatively inefficient. Results of these investigations indicated that marginal thermal gains didn't justify the additional weight of a heat exchanger. (A decrease in specific fuel consumption of 15 per cent resulted in an increase in weight of almost 100 per cent). The use of a heat exchanger was further complicated by the mechanical problems evidenced in the form of leakage losses. In connection with the Rover T.8 engine, does the British Admiralty intend to continue development work on a heat exchanger for this engine and if so, along what lines?

J. J. McMullen.[17] This paper represents actual proof that the gas turbine must be included in our plans for the future development of naval machinery. The actual installations and the operating experiences therefrom prove, without a doubt, that the gas turbine is now finding its various selected applications at sea.

The paper gives evidence that the British Admiralty conceived a gas-turbine program which would cover all possible applications. They then proceeded to accomplish these objectives on a very broad basis. After actual experience they soon realized that a few of their original ideas would not be successful and immediately diverted effort from these unsuccessful attempts into those which showed greater promise. This is the significant

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[16] Project Engineer, Gas Turbine Branch, Bureau of Ships, Department of the Navy, Washington, D. C. Assoc. Mem. ASME.

[17] Hudson Engineering Corporation, New York, N. Y.

point of the Admiralty's program. As a result, they are in their present position of having available at least one successful gas turbine for each of their original concepts.

The Admiralty deserves great praise for having conducted such an excellent program. Their program is most complete and inclusive with the exception of one type of gas turbine, namely, the closed-cycle type. Apparently the Admiralty feels that there is no future marine application for the closed-cycle gas turbine. If this is so the writer disagrees with their decision. Perhaps it is not necessary to include the closed cycle in the immediate program but, when one considers the future, especially for the utilization of nuclear power, the closed cycle represents a must for any long-range gas-turbine program. Even here, however, the British are fairly secure because John Brown in Clydebank is presently conducting a very progressive closed-cycle gas-turbine development.

W. A. Dolan, Jr.[18] The author is to be congratulated for the excellence of his paper on British naval gas turbines. The writer concurs, wholeheartedly, that the "greatest single advantage of the gas-turbine engine for naval applications is its ability to pack more power in less weight and space than any other prime mover." The U. S. Navy program is based largely on this same philosophy. Current test experience leads us to believe that reliability will be one of the major advantages of this machine aboard ship; however, this remains to be demonstrated completely.

In reference to the author's statements relative to the EL.60A type combustor, the writer would like to add that we have had excellent results with this type combustor on several of our programs. Test results indicate that low liner temperatures and relatively good combustion-chamber exit-temperature profiles are possible. For these reasons this type combustor is considered satisfactory for use at the higher temperatures required with a cooled gas turbine. These combustors also will be used in one of our newer engines, now under development. It is anticipated that this type combustor will result in considerable improvement in life of engine liner and nozzle parts.

The author refers to the possibility of inserting thermocouples in the turbine exhaust rather than in the turbine inlet of G2 engines. It has been our experience that there is less temperature stratification in the exhaust-gas stream than there is in the turbine-inlet gas stream. For this reason our engine specifications require that overtemperature alarm systems include thermocouples installed in the exhaust-gas stream. By mounting thermocouples in the exhaust, we obtain more accurate average gas-temperature readings.

The following questions are submitted for the author's comment:

(*a*) It is noted that kerosene was used to wash compressor blades on the Artouste engine. Does kerosene have any advantages over water?

(*b*) A trend toward journal-type bearings is apparent. Of what magnitude are the minimum losses anticipated with these type bearings? How will losses be kept to a minimum?

W. T. Sawyer.[19] The excellence of the paper makes it extremely difficult to add any constructive comment. The author's conclusion, from service experience, regarding the inadequacies of ball and roller bearings for long service in gas turbines is important. Is it not possible that the radial "shoe" or "pad" bearing may offer the required reliability which characterizes the sleeve bearing while at the same time affording more precise shaft positioning than is possible with the conventional sleeve bearing?

The importance of aerodynamic phenomena is stressed by the frequent references made to them in the paper. The effect of the struts at the exit from the turbine of G2, the interesting "rotating stall" found in the compressor of G2, the serious effect of poor entry to the compressor of the emergency gas-turbine set, and other less direct references, all illustrate the constant attention which must be given to aerodynamic detail.

Probably for the sake of brevity, the author omitted from his list of advantages of the gas-turbine-driven emergency generator its significantly greater ease of paralleling with steam-turbine generators, in comparison with diesel-driven equipment. Continuing the same comparison of gas-turbine versus diesel emergency generators, some specific figures on a modern 1000-kw installation in a carrier may be of interest for comparison with Appendix A. The actual installed weights will be as follows:

	Pounds per hp
Diesel engine, dry	21.8
Engine's portion of common bedplate	3.4
Fresh-water cooler	1.9
Muffler	1.2
Lube-oil cooler	0.9
Liquids (except fuel)	0.4
Air start tank	0.4
Lube-oil filter and strainer	0.1
Total specific weight	29.6

In addition, and supporting the author's comments regarding diesel maintenance, a weight of repair or maintenance parts for the engine amounting to 3.8 lb/hp will be carried on board! A complete spare gas-turbine engine could be carried for this weight alone.

It is noted that extensive running, modification, and development of gas-turbine engines are accomplished at the Admiralty Engineering Laboratory and at the Admiralty Test House at NGTE. The significance of the contributions of these Admiralty installations is evident from the text. It is my thought that fuller utilization of our U. S. Naval laboratories in a similar manner might permit more rapid and economical application of gas turbines, produced by our own industry for other fields, to effective marine service.

P. G. Tomalin.[20] The author has presented a valuable fund of information on gas turbines.

In the general classification, would the author in using the term "diesel fuel" compare the characteristics of that fuel with the American diesel fuel so that we would know if it is the same or different? The same question is advanced on residual fuels and particularly the question, was the residual fuel selected, and did it have any vanadium content?

Reference is made to life, under General Conclusions. Can it be assumed that design life is based on factor safety of 2, based on creep, or is some other criterion used?

The author refers to hull vibration and its effect on the engine. The term hull vibration is not understood since the action of the bearings would seem to indicate that it is caused by a torsional vibration which, in turn, results in a lateral vibration of the shafting from the propeller action.

In discussing the use of simple gas turbines as boost units in major warships, has any work been done in the United Kingdom on turbines suitable for combined duty? That is, turbines which

[18] Captain, USN, Bureau of Ships, Department of the Navy, Washington, D. C.

[19] Commander, USN, San Francisco Naval Shipyard, San Francisco, Calif.

[20] Principal Marine Engineer, United States Coast Guard Headquarters, Washington, D. C.

could be operated at high-temperature, low-pressure steam to increase the efficiency of the plant while cruising, utilizing waste heat from internal-combustion engines and using the same turbine as a boost turbine when supplying it with combustion gases from a combustion generator?

Reference is made to noise, giving the levels experienced and inferring that the noise generally is of high frequency. This does not seem to tally too well with the noise spectrum of aircraft jet-pipe noise, having large components of low-frequency noise. It will be difficult, if not impossible, to treat the intakes or exhaust to reduce the low-frequency noise by means of the material specified in the paper.

R. P. Ramsey.[21] This paper presents a refreshingly candid and thorough review of development and application problems on naval gas turbines. It is remarkably complete in its coverage and gives detailed answers to the universally asked question, "How do these schemes work out in practice?" It is a valuable contribution as contrasted with the usual technical presentation giving only predicted performance estimates. Appendix A is a valuable reference.

Certain naval requirements are ideally met by this new prime mover in its relatively simple, lightweight form. It is significant that the designs based on recent aircraft-turbine practices have passed their "teething troubles" more rapidly than the so-called heavy-duty types based on steam practice. It is clearly demonstrated that a gas turbine must be treated strictly as an aerodynamic machine, and that it is worth going to mechanical complexities if we can thereby eliminate a few feet of hot ducting. It also pays to design the compressors for broad range rather than maximum efficiency, to simplify matching of component and thus avoid delays which—at the current pace of developments—could well kill a project.

One surprising conclusion is that a long design life and/or high thermal efficiency is no longer directly associated with high unit weight. Weight in a combustion gas turbine seems to be affected primarily by the amount of regeneration, the number of turbine stages, and the amount of hot piping. All these can be major sources of trouble and are gradually being reduced on both short and long-life plants.

Naval service requirements differ from aircraft applications in that even a "short-life" plant still has to have long life under reduced loads, and thus is subject to vibration and fatigue loads on mechanical parts. This would explain the failure of aircraft-type ball-and-roller bearings in this service. The paper shows that where high specific output and economy are of importance, the more complex open-cycle turbines are preferable. The advantage of higher pressure ratios and higher temperatures are clearly indicated.

In considering the broad picture of applying gas turbines to naval propulsion, the free-piston gas-generator turbine presents many attractive features. Here the ultimate in high compression ratio and high combustion temperature can be realized while at the same time using a low-temperature turbine. Still, the resulting thermal efficiency obtained will be almost double that shown for any of the plants covered in the paper.

The problems of extra-large ducting, regeneration, and reheating are eliminated, and space requirements are at a minimum. Foundations are light and the over-all control of the plant is a simple, single control lever. Load response is practically instantaneous as compared with the longer time cycles required with the rotary compressor plant. Production costs are low.

The effects of ambient conditions are greatly minimized with free-piston machinery, and are more on the order of the variations experienced with diesel engines.

The emergency or auxiliary gas turbine is certainly a remarkable development and makes for a most straightforward example of design for application. It is doubtful that any other power plant heretofore developed has found as ready acceptance as these compact gas-turbine plants. Experience in this country with these units parallels that in Britain.

Author's Closure

Replying to Lt. Cdr. Fowden, the author confirmed that the required characteristics listed under General Conclusions in the paper were not in any order of importance. They were put together in the most convenient form to provide a connected narrative. It appeared doubtful whether complex-cycle gas turbines will have any future for combatant warships. Provided that there are no restrictions of weight and space, the complex cycle can be designed to give a higher efficiency than any other type, particularly at part load, but as soon as attempts are made to confine the complex-engine components within the small space necessary for most naval applications, the losses mount up and the efficiency consequently deteriorates. It is now possible to obtain nearly comparable efficiencies with a number of simple engines mechanically coupled together, and this arrangement is considerably lighter, simpler and less costly.

The prediction of gas-turbine efficiencies so far ahead as 1960 can be little more than a guess. Nevertheless, it is possible to make some sort of estimate based on the improvement which has taken place over the past years. Starting with the Gatric gas turbine in 1947 with a specific fuel consumption of 1.1 lb/hphr, the G.2 in 1951 had a specific fuel consumption of 0.8 lb/hphr. This trend can be followed in later designs and it would appear that by 1960 simple-cycle engines could be designed with specific fuel consumptions around 0.5 lb/hphr at full power.

Perhaps the paper had unduly emphasized "creep" as a basis for design life. The author merely meant to convey the impression that the life of gas turbines was based on high-temperature testing which is commonly called creep data. The method of calculating life which the British Admiralty advocated was to limit the stress in the blading or rotors to three quarters of the stress to rupture for the life of that particular engine, with the overriding consideration that the material did not enter an advanced stage of tertiary creep in that life. Further, in assessing this life the effect of thermal shock due to sudden variations in gas temperature was most important.

Mr. Hafer has very rightly commented on the extremely versatile nature of the gas-turbine engine. Considerable investigation and development undoubtedly are necessary to insure that the compressed air and exhaust gases are used to produce a combined naval installation with the best possible over-all efficiency. There were two reasons why the author considered that the inward radial-flow turbine fitted to the 200-hp Allen gas turbine was on the upper limit for size. (*a*) From the thermal-stress standpoint; any substantially larger inward radial turbines would necessarily have extremely high thermal stresses. (*b*) From the efficiency aspect; in larger sizes, the axial turbine could give a very much higher efficiency than the radial one. Of course there could never be any exact dividing line between axial and radial turbines, but, in general, it would appear that radial turbines had no advantages over axial units for a gas turbine of larger size other than the Allen 200 hp.

The particular cycle of the Allen 1000-kw gas turbine with a single-stage free-power turbine and heat exchanger is one of the most difficult from the governing aspect, and it would not be possible to obtain the same degree of close governing with the present design if the flywheel was removed. In particular, the

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instantaneous change from 0 to 25 per cent load would involve a substantially greater fall in speed. In a new design it probably would be possible to meet the governing specification without a flywheel but, to put the matter in its proper perspective, it can be seen from Fig. 30 that the flywheel makes no appreciable increase to the size of the Allen 1000-kw unit. Its weight is 500 lb which represents only 3 per cent of the total gas-turbine weight.

On the question of operating personnel, the gas-turbine installation should require substantially fewer numbers than a conventional steam-turbine plant. All the designs so far tried out in the British Navy are controlled by a single throttle lever. For a single gas turbine of the RM.60 type one skilled operator plus an additional rating to keep his eye on the gages is all that would strictly be necessary, but in a complete warship installation there were usually many auxiliaries which required additional operating personnel.

The author is indebted to Commander Shallenberg for giving details of the extremely interesting data on noise reduction and compartment ventilation which had been carried out at the Engineering Experiment Station. These data show clearly the very substantial reduction in noise which can be achieved in naval gas turbines when effective silencing is fitted.

In reply to the question on hydraulic couplings for reversing, this method is used in conjunction with the experimental 3500-hp marine gas turbine at Pametrada. The ahead drive is taken through a normal fluid coupling while astern power is derived from a hydraulic reversing coupling (torque converter) of the Föttinger type. This reverse gear has been successfully demonstrated on a brake in the Pametrada test house. It is interesting to note also that hydraulic reversing of this type was employed in certain German battleships at the time of the first world war.

Replying to Mr. Pitt, the author regrets that no noise measurements were taken of the Packard engines in MGB. 2009. Therefore he is not in a position to compare the noise of the Gatric gas turbine and the Packard engines. No specific trial to determine the effect of gas-turbine noise on human beings has yet been carried out by the British Admiralty but, without silencing, the high noise level would probably have a serious effect on operating personnel remaining in the same compartment. As mentioned in the paper, and supported by the figures quoted by Commander Shallenberg, it is comparatively easy to silence gas turbines down to an acceptable noise level. Moreover, gas turbines lend themselves readily to remote control, and operating personnel can be situated in a separate soundproofed control room. There is, therefore, no reason why noise should be a determining factor in preventing the extensive use of gas turbines in naval applications, but allowance must be made for the weight of silencing equipment required when comparing various installations.

The additional power loss involved in modifying the RM.60 to plain sleeve bearings and Kingsbury type thrusts would be about 300 hp at full power (5.6 per cent of total).

There is no doubt that single-shaft gas turbines can give superior regulation for generator applications, but they have three disadvantages compared with free-shaft engines which may be important in some naval applications.

(*a*) The part-load efficiency is lower, and naval generators operate for the majority of the time below full power.

(*b*) The single-shaft engine cannot withstand large momentary overloads or short circuits such as may occur under action conditions.

(*c*) The power required to start is considerably greater than in a free-shaft engine.

The author cannot quote any specific figures giving the effect of the thermal lag of the regenerator on governing, but it is a fact that the thermal capacity in a regenerator makes the governing problem more difficult. The Admiralty specify that gas-turbogenerators shall be capable of meeting sudden load changes up to 25 per cent with a speed fluctuation not exceeding $2^1/_2$ per cent, and take 100 per cent load changes on and off without stalling.

The author fully endorses Mr. Graves's view that the development of cooling methods should be given a high priority in any gas-turbine program. It is interesting to note that cooling developments in Great Britain are aimed primarily at increasing gas-turbine performance and not directly at reducing the quantities of critical materials in gas turbines, as appeared to be the case in the United States. However, it appeared that the greatly reduced weight and bulk resulting from the use of higher gas temperatures would, in fact, bring about a substantial decrease in the weight of critical materials per horsepower. Air cooling presented the simpler mechanical problems and development of this method was well advanced. Air already is being used today in many gas turbines for cooling turbine disks, blade roots, combustion chambers, and nozzle guide vanes, and several experimental engines are running with air-cooled blades. The use of air cooling, however, involves an appreciable loss to the cycle and, to obtain the highest thermal efficiencies, some form of liquid cooling was necessary. The British Admiralty had placed a contract with Pametrada to develop a liquid-cooled gas turbine but there are many problems yet to be overcome before this method can be put into service.

The British Admiralty are not at present sponsoring any research or development on heat exchangers for small gas turbines, but a keen interest is taken in the work being carried out by other concerns, both on conventional heat exchangers, rotary regenerators, and piston-type recuperators. So far as Great Britain is concerned, no really practical, lightweight, compact and durable heat exchanger with low leakage loss has yet been developed. If the attractive features of small gas turbines are to be maintained, a more satisfactory method of increasing the thermal efficiency may be to develop engines with higher pressure ratios and higher gas temperatures.

Mr. McMullen mentions the possibilities of the closed-cycle gas turbine. The author agrees that the closed cycle has some advantages for naval applications, in particular good part-load efficiency and reduced air consumption. But modern designs of open-cycle gas-turbine plants are so much lighter, smaller, and simpler, that their superiority for the majority of combatant warship applications can hardly be challenged. Moreover, when the development of blade-cooling techniques allows higher gas temperatures to be used in open-cycle gas turbines, this superiority will become even more marked, because of the difficulties of increasing the temperature in the closed cycle. Diversion of the limited resources of the British Admiralty to the development of closed-cycle plants does not therefore appear justified.

Replying to Captain Dolan, kerosene has been found superior for removing the fouling on compressors where the gas turbine is operating in an industrial atmosphere. Water will not remove the soot and grime deposits on compressor blades in a smoky atmosphere and the fouling at the Admiralty Engineering Laboratory was due to this cause and not to salt. In the open sea, fresh water has been found completely satisfactory for removing the salt deposits from compressors under all conditions of operation.

It is difficult to give a specific answer on the losses associated with journal-type bearings for gas turbines, since the magnitude varies with the type and size of engine, and particularly with the speed of rotation. However, it may be of interest to give two examples:

In the simple-cycle G.2 gas turbine of 4500 hp the change to journal-type bearings would involve an additional loss at full

power of about 40 hp (0.9 per cent of total power). In a complex cycle such as the RM.60, losses will be considerably higher. It is estimated that if the RM.60 were converted to journal-type bearings and Kingsbury thrusts, there would be an additional loss of about 300 hp at full power (5.6 per cent of total power). Moreover, the percentage loss at part load would be significantly greater since the H-P unit continues to run at high speed even at low power. In order to minimize the losses in journal bearings, it is essential to use small length/diameter ratios and to operate with oil at low viscosities. The losses in the Kingsbury type thrust bearings are even more serious than in journals and no really satisfactory method of reducing them has yet been evolved.

The author is indebted to Commander Sawyer for pointing out an additional advantage of gas-turbine-driven generators compared with diesel-driven equipment, viz., the significantly greater ease of paralleling with turbine generators. The very interesting table giving details of weights for modern 1000-kw diesel generators certainly illustrates the advantage in weight which the gas turbine possesses.

Radial "shoe" or "pad" type bearings appear to have advantages over the conventional sleeve bearings, but at present the British Admiralty have no operating experience with these novel bearings.

In answer to Mr. Tomalin, the diesel fuel mentioned in the paper is practically identical with American Navy diesel fuel and is very similar to commercial gas oil. The residual fuels chosen for testing were four Admiralty reference fuels, each of which was selected for its particular characteristics. All of these fuels contain vanadium in different proportions and one of them had a high concentration of both vanadium and sodium in the ash.

As stated earlier in the author's reply, the Admiralty advocate that the design life be based on three quarters of the stress required to fracture the material under the temperature and operating conditions obtaining in the gas turbine. This stress to rupture is based on the results of high-temperature test data.

The hull vibration mentioned was a hydrodynamic effect occurring at high speed when turning. The phenomenon has not been fully investigated to date, but appears to originate in the rudders which set up a disturbance causing the hull to vibrate in its fundamental mode. This results in violent vibration of the whole boat which, in turn, has damaged the gas-turbine bearings.

No work has yet been done in Great Britain on turbines suitable for combined duty of the type described by Mr. Tomalin. When comparing noise levels in aircraft gas-turbine jet pipes and naval gas-turbine exhausts, it must be remembered that there is a considerable difference between the two systems. In naval gas turbines, the exhaust gases are traveling at much lower velocities and there are usually free output turbines, volutes, and even heat exchangers between the turbine outlet and the exhaust. There has so far been no difficulty in keeping the exhaust noise on Admiralty gas-turbine installations down to an acceptable level, and judging by the contribution received from Commander Shallenberg, this had been the case in the U. S. Navy also.

The author agrees with Mr. Ramsey that the air-piston gas generator appears to offer many advantages for marine propulsion. Nevertheless, the engines built to date are heavy and bulky compared with simple gas turbines. The only sets of which the author has detailed firsthand knowledge are those developed in France which have specific weights of nearly 40 lb/hp but, on the other hand, have obtained high thermal efficiencies of about 35 per cent. These engines are probably more suited to merchant-marine propulsion than to combatant warships where weight and space considerations are so vital.

In conclusion, the author wishes to thank all those who have taken part in this most stimulating discussion on his paper. He assures them that their contributions will be of great assistance in determining the future application of gas turbines to naval vessels.

The Dynamics of Filled Temperature-Measuring Systems[1]

By OTTO MULLER-GIRARD,[2] ROCHESTER, N. Y.

The response equations of a two-time-constant interacting network as applied to liquid-filled tube systems are developed. The solutions to a unit step, ramp, and sinusoidal temperature input are given. Some practical aspects of response work based on experiments with mercury-filled thermal systems are presented and a few recommendations concerning the specification of the speed of response are included.

Introduction

LIQUID-filled tube systems are used to determine the temperature of fluids in industrial processes. Temperature controlled processes are closed-loop systems in which the difference between the desired temperature and the measured temperature is used to effect the control of the process. A filled-tube system used to sense the temperature of the process constitutes part of the feedback loop and as such its dynamic characteristics affect the stability and the attainable quality of control of the process. Liquid-filled systems, in general, and mercury-filled systems in particular, when subjected to a sudden change in temperature, exhibit a "dip" or an initial indicated temperature in opposition to the applied temperature. This dip, its cause and effect on the frequency response, are of particular interest here.

Theory

Fig. 1 shows the physical configuration of a filled system. As long as the bulb and its environment are at the same temperature, no heat (thermal energy) will flow between the bulb and its surroundings. However, when the environmental temperature changes to a new value, heat will flow into or out of the bulb until enough heat has been stored or released to make the bulb and external temperature the same. Heat may be stored in both the shell and the filling liquid. In order for heat to reach the shell (an up-step in temperature is assumed here) the heat must diffuse through the outside fluid film and perhaps part of the shell. A part of this heat will be stored due to the heat capacity of the shell, raising the temperature of the shell. Since initially little heat has penetrated beyond the shell, the temperature increase of the shell causes it to expand, and thus increase the internal volume. An increase in shell volume causes liquid to flow from the Bourdon into the bulb. The decrease in fluid in the Bourdon causes the Bourdon to contract and indicate a temperature lower than the temperature prior to the up-step.

The heat which is not stored in the shell eventually will diffuse into the filling liquid and raise its temperature. With the volumetric temperature coefficient of expansion of the filling liquid being normally much larger than for the shell material, a net increase in liquid volume results, causing a reversal of motion and an eventual true indication of the new temperature of the external fluid.

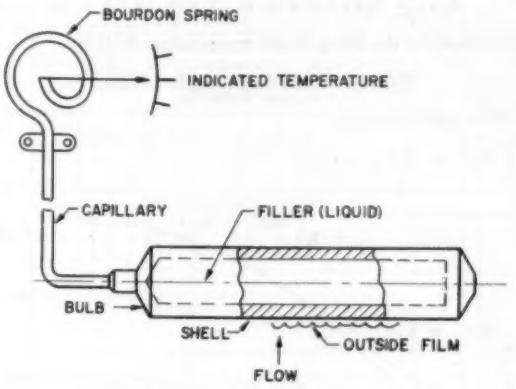
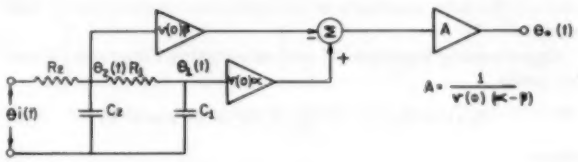


Fig. 1 The Tube System



θ_i = TEMPERATURE INPUT (FORCING FUNCTION)
θ_1 = FILLING LIQUID TEMPERATURE
θ_2 = SHELL TEMPERATURE
θ_0 = INDICATED TEMPERATURE
R_1 = INTERNAL RESISTANCE
R_2 = INPUT RESISTANCE
C_1 = FILLING LIQUID HEAT CAPACITY
C_2 = SHELL HEAT CAPACITY
$v(o)$ = VOLUME OF BULB PRIOR TO A CHANGE IN TEMPERATURE
α = VOLUMETRIC COEFFICIENT OF EXPANSION OF FILLING LIQUID
β = VOLUMETRIC COEFFICIENT OF EXPANSION OF THE SHELL MATERIAL

Fig. 2 Analogy to the Tube System

From the foregoing, the analogy shown in Fig. 2 becomes apparent. The indicated temperature of the system as a function of time is

$$\theta_0(t) = A[v(0)\alpha\theta_1(t) - v(0)\beta\theta_2(t)] \quad \ldots\ldots\ldots [1]$$

where A is a scale factor and the remaining terms are defined in Fig. 2. The assumption here is that the outside film coefficient and the coefficients of expansion, the thermal conductivity, and the specific heat of the shell and filling fluid remain constant during a transient, making the performance of the system independent of the starting level which is arbitrarily taken as zero. If the tube system is calibrated such that the scale reading cor-

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[1] The material contained in this paper was used by the author in partial fulfillment of the requirements for an MS degree at the University of Rochester, Rochester, N. Y., 1954. The work was undertaken in co-operation with the ASME-IRD Application Committee, Project AP-14.

[2] Research Engineer, Taylor Instrument Companies.

Contributed by the Instruments and Regulators Division and presented at the Semi-Annual Meeting, Pittsburgh, Pa., June 20–24, 1954, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, March 30, 1954. Paper No. 54—SA-29.

responds to the temperature of the external fluid in the steady state, then Equation [1] becomes

$$\theta_0(t) = \frac{1}{(\alpha - \beta)} [\alpha\theta_1(t) - \beta\theta_2(t)] \quad [2]$$

The transformed solution of the shell temperature $\theta_2(s)$ where s is the Laplace operator, in terms of any temperature input $\theta_i(s)$ is

$$\theta_2(s) = \theta_i(s)(R_1C_1s + 1)/(T_1s + 1)(T_2s + 1) \quad [3]$$

The solution for the filling-liquid temperature $\theta_1(s)$ is

$$\theta_1(s) = \theta_i(s)/(T_1s + 1)(T_2s + 1) \quad [4]$$

where for both solutions

$$T_1 = \frac{R_1C_1 + R_2(C_1 + C_2)}{2} + \sqrt{\left[\frac{R_1C_1 + R_2(C_1 + C_2)}{2}\right]^2 - R_1C_1R_2C_2}$$

and

$$T_2 = \frac{R_1C_1 + R_2(C_1 + C_2)}{2} - \sqrt{\left[\frac{R_1C_1 + R_2(C_1 + C_2)}{2}\right]^2 - R_1C_1R_2C_2}$$

and are the time constants of an equivalent noninteracting network.

Transforming Equation [2] and substituting Equations [3] and [4] yields

$$\theta_0(s) = \theta_i(s)(1 - T_3s)/(T_1s + 1)(T_2s + 1) \quad [5]$$

where

$$T_3 = \frac{\beta R_1C_1}{\alpha - \beta} = K(T_1 - T_2) - T_1$$

Equation [5] is the general transformed solution of the indicated temperature to a forcing function as yet unspecified.

It must be mentioned here that the proposed analogy to the tube system is an approximation, for the resistances and the heat capacities are distributed throughout the bulb and are not physically lumped as assumed in the analogy. However, the analogy may be justified on the basis that it is simple while still conforming closely to the observed behavior. Also, in the treatment of the analogy, the resistances and capacities are assumed to be independent of temperature during a change, yielding a solution the form of which is not affected by the amplitude of the input. The effects of these assumptions on the response will be treated later on.

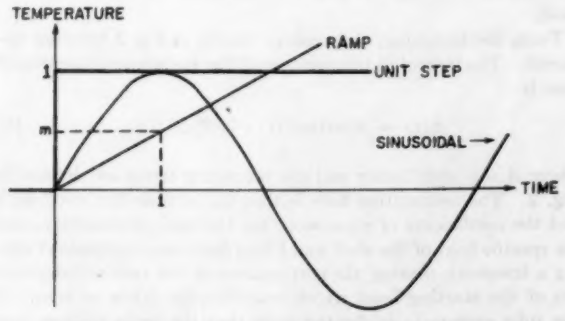

FIG. 3 VARIOUS TYPES OF TEMPERATURE INPUTS

Three types of input which have been produced experimentally will be considered. The first is the unit step or sudden change of temperature; the second, the temperature ramp; and the third, the sinusoidally varying temperature of variable frequency. All three inputs are shown in Fig. 3.

THE UNIT STEP

The transform of the unit step is

$$\theta_i(s) = \frac{1}{s} \quad [6]$$

Applying this to the general solution of the analog yields

$$\theta_0(s) = (1 - T_3s)/s(1 + T_1s)(1 + T_2s) \quad [7]$$

Performing the inverse transformation gives the response of the system as a function of time; thus

$$\theta_0(t) = 1 - Ke^{-t/T_1} + (K - 1)e^{-t/T_2} \quad [8]$$

where

$$K = (T_1 + T_3)/(T_1 - T_2)$$

Rewriting Equation [8] to give the incomplete response which is the difference between the final value of the response and the instantaneous response yields

$$1 - \theta_0(t) = Ke^{-t/T_1} - (K - 1)e^{-t/T_2} \quad [9]$$

The composition of the incomplete response is shown in Fig. 4. If T_1 is greater than T_2 by a factor of about 3 or more, a plot of the

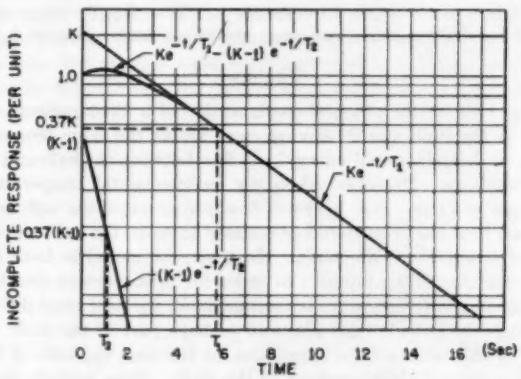

FIG. 4 INCOMPLETE-RESPONSE COMPOSITION

log of the incomplete response versus linear time may be broken up graphically into straight lines, the slopes of which are inversely proportional to the time constants and the intercepts of which are K and $(K - 1)$. It is this feature of being able to separate the constants from the incomplete response data which makes the step response such a valuable experimental method.

In order for the two time-constant systems to exhibit a dip, the slope of the response to a step at "zero plus" time must be in opposition to the applied step. The condition for a dip for systems conforming to the proposed analogy is that $K > T_1/(T_1 - T_2)$ or that $\beta > 0$ when α, R_1, and C_1 are greater than zero.

TEMPERATURE RAMP

The temperature ramp has been used recently to determine a characteristic of temperature-sensing elements.[3] The equation

[3] "Time-Lags in Thermometer Pockets," by W. Fishwick, Trans. Society of Instrument Technology, vol. 4, March, 1952.

for the temperature ramp is of the form

$$\theta_i(t) = \theta_i(0) + mt \quad \text{for } t \geq 0$$

where m is the change in temperature per unit time (t) or the slope of the ramp.

The response of a filled system after the transient has died out is

$$\theta_0(t) = \theta_i(0) + m(t - T_1 - T_2 - T_3) \text{ for } t \gg T_1 + T_2 + T_3 \quad . . [10]$$

From this it may be seen that the indicated temperature lags the input temperature by the sum of the time constant, thus

$$\text{Lag } (L) = T_1 + T_2 + T_3$$

The concept of the constantly rising temperature input is a useful one since many heating or cooling processes exhibit such a characteristic. The knowledge of the lag (L) allows the temperature to be evaluated. However, for automatic-control purposes the separation of the time constants is necessary in order to arrive at the frequency response of the system.

Frequency Response

Assuming, as previously, that the filled-tube system may be represented by the interacting "RC" network, then the steady-state response to a sinusoidal temperature variation of amplitude (A) as derived from Equation [5] is

$$\theta_0(t) = A \sqrt{\frac{1 + (\omega T_3)^2}{[1 + (\omega T_1)^2][1 + (\omega T_2)^2]}} \sin(\omega t + \psi) \quad . [11]$$

where

$$\psi = \tan^{-1}(-\omega T_3) - \tan^{-1} \omega T_1 - \tan^{-1} \omega T_2$$

Both the magnitude ratio (the square-rooted expression) and the phase (ψ) are frequency dependent. In the upper limit, that is, when ω approaches infinity, the magnitude ratio becomes inversely proportional to the frequency and the phase approaches a final value of minus 270 deg. On a log-magnitude ratio versus log-frequency plot, the magnitude-ratio curve assumes a final slope of minus 1.

It may be noted here that Equation [5] or [11] describes the performance of a nonminimum phase network. A nonminimum phase network may be defined, for the purpose of this paper, as a network for which the magnitude ratio versus frequency characteristic does not define uniquely the phase versus frequency characteristic. The nonminimum phase behavior results from the action of the shell, through its coefficient of expansion, on the response of the system. Earlier it was mentioned that the expansion of the shell caused the dip in the response to a step and from this it may be concluded that any system exhibiting a tendency to dip will behave like a nonminimum-phase network.

A typical frequency-response plot for a mercury system is shown in Fig. 5 where curve A represents experimental data and curve B the response calculated by Equation [11].

Resistances and Capacities

Up to this point no effort has been made to show how the resistances and capacities in the analogy may be computed. Even though no general solution can be offered, the following definitions have given good agreement with step and frequency-response data for a set of cylindrical mercury-filled-tube systems.

The internal resistance R_1 for mercury systems is

$$R_1 = \frac{\ln \sqrt{2}}{k_{\text{filler}}} + \frac{\ln \sqrt{(D_o^2 + D_i^2)/2D_i^2}}{k_{\text{shell}}}$$

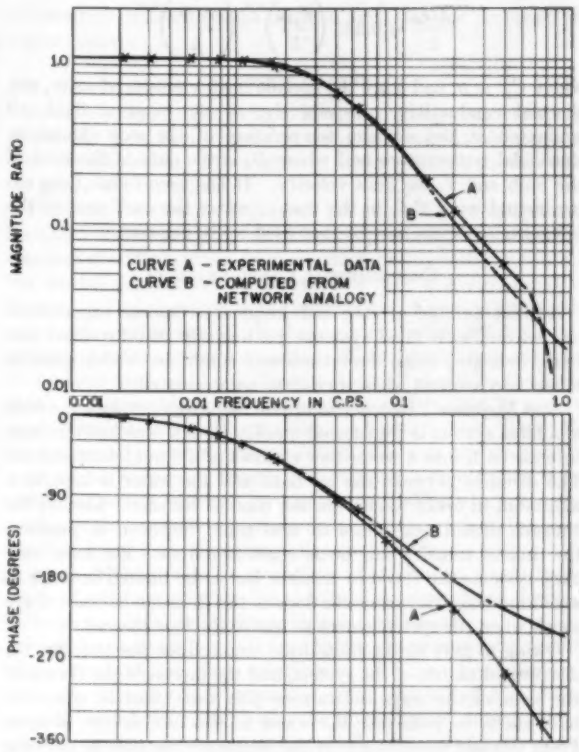

Fig. 5 Frequency Response of Mercury-Filled Tube System

The input resistance R_2 is

$$R_2 = \frac{\ln \sqrt{2D_o^2/(D_o^2 + D_i^2)}}{k_{\text{shell}}} + \frac{2}{D_o h}$$

For both equations D_o is the outside diameter, D_i the inside diameter of the shell, k_{filler} and k_{shell} are the thermal conductivities as designated, ln is the natural logarithm (base e), and h the outside film coefficient as defined later. In the foregoing definitions the length of the bulb is assumed to be $1/2\pi$ units and the capacities of the shell and filling fluid are lumped at diameters which split the distributed shell and filling-liquid capacities, respectively, in halves.

The filling-liquid capacity C_1 is

$$C_1 = \rho_1 C_{p1} D_i^2/8$$

and the shell capacity is

$$C_2 = \rho_2 C_{p2}(D_o^2 - D_i^2)/8$$

where $\rho_{1,2}$ and $C_{p1,2}$ are the density and specific heat, respectively, of the filling liquid and the shell. Again, a bulb length of $1/2\pi$ units is assumed. In the response equations R and C always occur in combination. With the external flow being perpendicular to the bulb, the RC product is independent of length providing the bulb is sufficiently long to make end conditions negligible. A length of $1/2\pi$ units is used to simplify the numerical work.

For evaluating the outside-film coefficient in any fluid with Reynolds number greater than 100 and with normal flow, the following form of Nusselt's equation as given by Stoever[4] is recommended

[4] "Applied Heat Transmission," by H. J. Stoever, McGraw-Hill Book Company, Inc., New York, N. Y., 1941.

$$\frac{hD_o}{k} = 0.385 \left(\frac{C_p\mu}{k}\right)^{0.3} \left(\frac{\rho V D_o}{\mu}\right)^{0.54}$$

where C_p, μ, ρ, and k are the specific heat, viscosity, density, and thermal conductivity, respectively, of the external fluid, all evaluated at the average temperature of the step change or sinusoidal temperature and where D_o is the outside diameter of the bulb and V the fluid velocity. It has been found from experimental work that in the case of water the coefficient in the foregoing equation may be increased by 10 per cent.

Experimental Considerations

A great number of step and frequency-response experiments carried out with filled systems, particularly mercury-filled systems, indicated some limitations and conditions which must be taken into account when such data are evaluated.

Step Response. In order to obtain step-response data the bulb of a filled system is transferred quickly from a constant-temperature air bath into a water-flow channel of different temperature. The distance between the air bath and the water is kept to a minimum in order to shorten the time of transfer. Ideally, the transfer should take place in zero time; however, in practice, the time of transfer may be as much as 0.3 sec. For some mercury systems the speed of transfer has to be limited in order to avoid inducing violent oscillations in the Bourdon because of the shock when the system comes to rest in the flow channel.

Owing in part to the finite time required for the transfer, the distributed nature of the system, and the inertia of the Bourdon, the slope of the response at zero plus time, that is, when the bulb starts to penetrate the water in the test section, is zero. From this one may infer that the frequency response of the tube system will show a higher rate of attenuation at the higher frequencies than the proposed analogy. Frequency-response experiments show this to be true.

For all systems tested by step response, the data when plotted as log of incomplete response versus linear time could be broken up readily into two straight lines, yielding two time constants and the coefficient. In some instances the data could be broken up into three straight lines; however, the third time constant thus obtained is of the order of the timing uncertainty and may be neglected. In addition to the thermal lags, some systems exhibit a capillary lag[5] which, if the capillary is long, may be larger than the smaller of the two thermal time constants.

Under transient conditions, the temperature gradient during the initial part of the response is steep and the expansion of the shell may be affected by thermal stresses in the shell. The response also is affected by transient pressures inside the shell which tend to expand the shell and compress the filling liquid. Pressure transients are a direct consequence of the capillary lag which retards the fluid flow between bulb and Bourdon.

When the amplitude of the temperature step is large, about 30 deg F or more, the value of the outside-film resistance will change markedly during the transient and affect the response. Large step amplitudes cause the normally straight portions of the log incomplete response plot to assume a curvature which makes the determination of the time constant difficult.

Despite the mentioned complicating factors, the response of mercury-filled-tube systems with 10-ft capillaries, when subjected to a 10 deg F step, is well accounted for by the two-time-constant analogy.

Step-response experiments are relatively simple to carry out and do not require too much time and equipment. The analysis of the step-response data, though it is time-consuming, yields a set of numbers which fixes the performance of the system and which may be correlated readily with the physical constants of the systems. It is this last feature which makes the step-response experiments so desirable.

Frequency Response. Frequency-response experiments on filled-tube systems are not difficult to carry out provided one has a thermal sine-wave generator[6] at hand. However, a thermal sine-wave generator is expensive and much time is required to run a frequency response for a filled-tube system. The frequencies used are very low, from about 0.001 to 0.75 cycle per sec. In order to insure a fair representation of the system, from 2 to 20 complete cycles, depending on the frequency, must be allowed to occur after each frequency change to permit transients caused by the change to die out before data are taken. The recorded input and output temperatures then may be converted readily into a frequency-response plot. This plot may be used directly to solve an automatic-control problem, provided the fluid conditions of the frequency-response plot correspond to the condition of the problem. Translating a frequency-response plot from one fluid condition to another is difficult since such a plot may not be readily divisible into its constituent transfer lags.

Conclusions and Recommendations

For the purpose of studying the dynamics of liquid-filled-tube systems a simple two-time-constant interacting-network analogy was presented. A study of mercury-filled systems indicates the simple presentation to give close correspondence with the observed behavior in the region of interest from an automatic control point of view, that is, for phase lags less than about 60 angular deg. Since at the present, data are not available on systems other than the mercury-filled systems, a study of such systems seems desirable. For some time the specification of the speed of response of filled systems has been of some concern to the user as well as the manufacturers of such systems. At present, the informally but widely used method of specifying the speed of response of temperature-measuring systems is the 63 per cent response time to a step input, with the external fluid conditions often vaguely specified as "well-agitated water." The 63 per cent response time is meaningless when applied to systems exhibiting more than a single lag. Because of the ambiguity of the present method for specifying the dynamics of tube systems, some suggestions will now be made as to how the speed of response of mercury-filled systems in particular, may be specified. The hope is that as more information on temperature-sensing elements becomes available a uniform standard for specifying the dynamics of such systems can be established.

In view of the fact that to a great number of users the frequency-response information is of primary concern, one method of specifying the characteristics of a filled system might be to supply a plot of magnitude ratio and phase versus frequency at the desired fluid condition, or at some standard condition such as water flowing normal to the bulb at 1 fps at a temperature of 100 F. Although such a specification would yield all the necessary information for one fluid condition, it would be difficult to translate the information to suit other fluid conditions.

Another possible method based on frequency-response data may be to specify the frequency and phase lag at a magnitude ratio of 0.707, at a given fluid condition. This method, although yielding little quantitative information, could be very useful in comparing systems at the same fluid condition. The system with the highest frequency and the least phase lag is the most desirable one from an automatic-control point of view.

Finally, the least ambiguous method of specifying the charac-

[5] "Fundamentals of Automatic Control," by G. H. Farrington, Chapman & Hall, Ltd., London, England, 1951.

[6] "The Design and Operation of a Thermal Sine Wave Generator," by R. Looney, thesis in preparation at the University of Rochester, Rochester, N. Y., 1954.

teristics of two time-constant (mercury) systems, is to specify the two equivalent noninteracting time constants, the coefficient K, the volumetric coefficients of expansion of the shell and filling fluid, and the external fluid condition at which the time constants were measured. Such a specification, provided the fluid condition is defined rigidly, would allow the user to calculate the transient or frequency-response characteristics of the system with fair accuracy for any fluid condition.

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Discussion

Yasundo Takahashi.[7] The author's contribution in wiping out the ambiguities of dynamic property of temperature measuring systems should be highly regarded. In thermal processes, film coefficients are quite changeable. Unfortunately, the input resistance, hence the outside film coefficient, is included in all time constants T_1 to T_3. This is the reason why each frequency-response datum is applicable only for a rigidly defined pattern of fluid condition. Isn't there any short-cut method to get rid of this limitation? For example, let the frequency response of a system under one fluid condition be completely given. Could not one translate this to another fluid condition, given only limited and easily determinable information on the latter, such as maximum reaction rate for a unit step input, or something like that?

Also, the writer would like to know the fault as compared with the author's method, of the conventional approximate representation of the system by a dead-time plus single-capacity response, replacing the dip by dead time, if unit step test result is available, and if frequency response for bandwidth having phase lags less than about 60 angular deg is of primary importance?

A few years ago the writer had an opportunity to discuss with Mr. J. G. Ziegler a process having a dip in step response. From this experience the writer believes that the very clear relations pointed out in Fig. 4, between three time constants which predominate the process behavior, are important not only for the temperature-measuring tube system, but also for processes of similar natures.

[7] Professor of Mechanical Engineering, Institute of Industrial Science, University of Tokyo, Japan. Visiting Fellow of M.I.T.

Author's Closure

Because of service in the U. S. Army the author was unable to attend the 1954 Semi-Annual Meeting and present his paper or hear the discussion. He wishes to thank Mr. W. I. Caldwell for presenting the paper.

Professor Takahashi's interest in the paper is appreciated and his question concerning the short cut in translating the frequency response of a filled system from one medium to another, when for the second only the maximum reaction rate to a step input is known, may be answered in the affirmative. Knowing the maximum reaction rate fixes T_1 for those systems which exhibit a fair separation between time constants. With mercury-tube systems over a large range of film coefficients it can be shown on the basis of the RC-network analogy or experimental data that T_2 and T_3 are relatively unaffected by the film coefficient. Also, T_2 and T_3 are numerically similar, offsetting their effects on the magnitude ratio versus frequency graph, and yielding a magnitude ratio versus frequency response for the whole system which is very similar to the response of a single time constant system, the time constant being T_1. Thus translating the magnitude ratio graph to a new medium consists essentially of drawing a single time-constant response graph using the time constant T_1 as derived from the maximum reaction rate in the new medium.

Since the original T_1 is well fixed by the magnitude ratio versus frequency graph, the phase contribution of T_1 may be subtracted from the original phase versus frequency graph and the phase contribution of the new T_1 may be added in its proper place, resulting in the phase versus frequency response for the new condition. This procedure gives quite useful results though it must be remembered that there are approximations beyond those made in the RC-network analogy.

Other useful approximations are that for large film coefficients the response becomes independent of the film coefficient, whereas for small film coefficients the system behaves as a single time-constant system, the time constant being $\frac{2}{D_0 h}(C_1 + C_2)$.

The fault to be found with the dead-time plus single-capacity response approximation is that this approximation yields a larger phase lag than is found experimentally or with the RC-network analogy, even for a band width limited to 60-deg phase lag.